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SIMULATION OF SOLAR HEATING AND COOLING SYSTEMS USING THE CONTINUOUS SYSTEM MODELING PROGRAM

by

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NOMENCLATURE

a	absorption coefficient, expressed as decimal of the impinging solar radiation that is absorbed by the wall surface
A	house total outside surface, ft^2
A_c	collector area, ft^2
A_{roof}	area of roof, ft^2
A_w	area of window, ft^2
A_{wall}	area of wall, ft^2
C	a function of collector tilted angle, defined in Eq.(9)
C_a	air change rate, per hour
C_{ai}	the capital cost of an air conditioner, \$
C_b	bond conductance, Btu $hr^{-1} ft^{-1} {}^oF^{-1}$ (see Figure 3.2)
C_c	the capital cost per unit area of collector, \$/ft^2
C_{CON}	the annual cost of the conventional system, \$
C_e	the capital cost of additional equipment for a solar system, \$
C_f	the capital cost of an electric furnace, \$
C_h	heat pump capacity, Btu hr^{-1}
C_{hp}	the capital cost of a heat pump, \$
C_m	the annual cost of maintenance, \$
C_{MA}	the annual cost for MODEL A, \$
C_{MB}	the annual cost for MODEL B, \$
C.O.P.	coefficient of performance, defined in Eq.(47)
$C.O.P._I$	ideal coefficient of performance, defined in Eq.(46)
C_r	the unit cost of power, \$/KWh
C_{st}	the capital cost of storage(medium, container, and insulation), \$
C_w	the capital cost of an electric hot water heater, \$

D	tube diameter, ft
D_i	tube inside diameter, ft
$(DS)_A$	the annual dollar savings for MODEL A, \$
$(DS)_B$	the annual dollar savings for MODEL B, \$
f	a function of h_w and N, defined in Eq.(10)
F	fin efficiency, defined in Eq.(4)
F'	collector efficiency factor, defined in Eq.(3)
F_R	collector heat removal factor, defined in Eq.(2)
F_w	the dimensionless ratio of the solar heat gains to the incident solar radiation
F_1	defined in Eq.(14)
F_2	defined in Eq.(15) and Eq.(45)
F_3	defined in Eq.(16)
F_4	defined in Eq.(61)
F_5	defined in Eq.(62)
$h_{f,i}$	heat transfer coefficient between the fluid and the tube wall, Btu $hr^{-1}ft^{-2}{}^{o}F^{-1}$
h_w	wind heat transfer coefficient, Btu $hr^{-1}ft^{-2}{}^{o}F^{-1}$
H	local solar time, expressed as the hour angle
H_I	heat gain due to infiltration and ventilation, Btu hr^{-1}
H_{roof}	heat gain through the roof, Btu hr^{-1}
$H_{r,wall}$	heat gain through a wall due to solar radiation, Btu hr^{-1}
$H_{t,wall}$	heat gain through a wall due to air temperature difference, Btu hr^{-1}
H_w	heat gain through a window, Btu hr^{-1}
H_{wall}	heat gain through a wall, Btu hr^{-1}
I_a	actual intensity of the solar radiation striking the surface, Btu $hr^{-1}ft^{-2}$

I_c	actual solar radiation impinging on the collector, Btu $hr^{-1}ft^{-2}$
I_M	measured solar radiation on horizontal plain, Btu $hr^{-1}ft^{-2}$
I_n , I'_n	annual factor of investment
k	fin thermal conductivity, Btu $hr^{-1}ft^{-1}{}^{o}F^{-1}$
L	local latitude, 39° for Manhattan
m	defined in Eq.(5)
$(\dot{m}c_p)_c$	collector fluid capacitance rate, Btu $hr^{-1}{}^{o}F^{-1}$
$(\dot{m}c_p)_{CL}$	cooling load fluid capacitance rate, Btu $hr^{-1}{}^{o}F^{-1}$
$(\dot{m}c_p)_{CH}$	heat pump cooling fluid capacitance rate, Btu $hr^{-1}{}^{o}F^{-1}$
$(\dot{m}c_p)_{HL}$	house heating fluid capacitance rate, Btu $hr^{-1}{}^{o}F^{-1}$
$(\dot{m}c_p)_{LW}$	hot water heating fluid capacitance rate, Btu $hr^{-1}{}^{o}F^{-1}$
$(mc_p)_{s,1}$	heat capacity of the storage tank section 1, Btu ${}^{o}F^{-1}$
$(mc_p)_{s,2}$	heat capacity of the storage tank section 2, Btu ${}^{o}F^{-1}$
$(mc_p)_{s,3}$	heat capacity of the storage tank section 3, Btu ${}^{o}F^{-1}$
N	number of glass cover
P	portion, expressed as a decimal, of the absorbed solar radiation that is transmitted to the inside
$P_{A,c}$	the annual power requirements for cooling for MODEL A, KWh
$P_{A,h}$	the annual power requirements for heating for MODEL A, KWh
$P_{B,c}$	the annual power requirements for cooling for MODEL B, KWh
$P_{B,h}$	the annual power requirements for heating for MODEL B, KWh
$P_{CON,c}$	the annual power requirements for cooling for the conventional system, KWh
$P_{CON,h}$	the annual power requirements for heating for the conventional system, KWh
Q_{AUW}	hourly auxiliary energy required for hot water heating, Btu hr^{-1}
$\overline{Q}_{AUW}$	long-term cumulative auxiliary energy required for hot water heating, Btu hr^{-1}

Q_{AUX}	hourly auxiliary energy required for house heating, Btu hr^{-1}
$\overline{Q}_{AUX}$	long-term cumulative auxiliary energy required for house heating, Btu hr^{-1}
Q_{BS}	hot water heating load supplied by the solar system, Btu hr^{-1}
Q_{HL}	the house heating load, Btu hr^{-1}
Q_u	rate of the collector energy collection, Btu hr^{-1}
Q_{WL}	the hot water heating load, Btu hr^{-1}
t	time, hour
T_a	ambient temperature, ^oF
T'_a	ambient temperature, ^oR
T_B	temperature of the basement, ^oF
T_{CH}	temperature of water returning from the heat pump in cooling operation, ^oF
T_{CL}	temperature of water returning from the cooling load, ^oF
T_{hot}	temperature of the heated domestic water leaving the hot water heat exchanger, ^oF
T_i	indoor design temperature, ^oF
$T_{f,i}$	temperature of the fluid at the collector inlet, ^oF
$T_{f,o}$	temperature of the fluid at the collector outlet, ^oF
T_{Lh}	temperature of water returning from the heat pump in heating operation, ^oF
T_{Lr}	temperature of water returning from the heating load, ^oF
T_{LW}	temperature of water returning from the hot water heat exchanger, ^oF
T_p	plate temperature, ^oF
T'_p	plate temperature, ^oR
$T_{s,1}$	temperature of water in section 1 of the storage tank, ^oF
$T_{s,2}$	temperature of water in section 2 of the storage tank, ^oF

$T_{s,3}$	temperature of water in section 3 of the storage tank, ^oF
T_1	heat pump suction temperature, ^oR
T_2	heat pump condensing temperature, ^oR
U	house equivalent heat loss coefficient, Btu $hr^{-1}ft^{-2}{}^oF^{-1}$
$(UA)_{s,1}$	heat loss coefficient of the storage tank section 1, Btu $hr^{-1}{}^oF^{-1}$
$(UA)_{s,2}$	heat loss coefficient of the storage tank section 2, Btu $hr^{-1}{}^oF^{-1}$
$(UA)_{s,3}$	heat loss coefficient of the storage tank section 3, Btu $hr^{-1}{}^oF^{-1}$
U_{be}	collector back insulation heat loss coefficient, Btu $hr^{-1}ft^{-2}{}^oF^{-1}$
U_L	heat loss coefficient of the collector, Btu $hr^{-1}ft^{-2}{}^oF^{-1}$
U_{roof}	heat loss coefficient of the roof, Btu $hr^{-1}ft^{-2}{}^oF^{-1}$
U_w	heat loss coefficient of a window, Btu $hr^{-1}ft^{-2}{}^oF^{-1}$
U_{wall}	heat loss coefficient of a wall, Btu $hr^{-1}ft^{-2}{}^oF^{-1}$
V	wind speed, mile hr^{-1}
V_o	house total air-conditioned volume, ft^3
W	the distance between the tubes, ft
β	solar attitude above the horizontal, degrees
γ	wall-solar azimuth, degrees
δ	solar declination, degrees
δ'	fin thickness, ft
ε	collector tilted angle from horizontal, degrees
ε_g	emittance of glass, 0.88
ε_p	emittance of plate
ϕ	solar azimuth measured from the south, degrees

θ	the incident angle, degrees
θ_H	the incident angle on a horizontal plane, degrees
θ_v	the incident angle on a vertical plane, degrees
$\theta_{49°}$	the incident angle on the 49° tilted collector surface, degrees
$(\tau\alpha)$	transmittance-absorptance product of cover system
σ	Stefan-Boltzmann constant, 0.1712×10^{-8} Btu $hr^{-1} ft^{-2} {}^{o}R^{-4}$

CHAPTER 1. INTRODUCTION

During the past few years, there has been a growing awareness in the world of the need for alternate sources of primary energy. Many studies have shown us that the world's fossil fuel resources, which might be considered a form of stored solar energy, will soon be depleted in meeting man's enormous appetite for energy. Wilson and his colleagues [1] conclude that the supply of oil will fail to meet increasing demand before the year 2000, most probably between 1985 and 1995, even if energy prices rise 50% above current levels in real terms. The task for the world will be to manage a transition from dependence on oil to greater reliance on other fossil fuels, nuclear energy and, later, renewable resources of energy - e.g., solar,wind-power, and wave power.

Evidently, the renewable resources, especially solar, are likely to become increasingly important in the 21th and later centuries. Such perceptions lead us naturally to consider the ways of collecting, storing, and using solar energy.

The use of solar energy is not a new revelation. Since time immemorial, priests ignited temple fires by concentrating solar energy, thus adding mystery to their rituals. According to Bereny [2], the first major event in solar energy history was in 1772 when a French chemist, Lavoisier, generated temperatures up to 1,750°F by using two hollow glass lenses filled with white wine to focus sunlight. Then in 1870's, John Ericsson developed solar engines. He found that they could not compete with fuels of that time. After that, in the late 19th century and early 20th century, many solar-assisted instruments were designed and constructed. These

included solar stills, solar engines, solar cooking stoves, solar heating collectors, and solar batteries.

In recent years, research and development on direct use of solar energy has led to a much better understanding of the principles of construction of solar systems. These solar systems might be classified as: (A) Solar Energy for Water Heating and Steam Generation; (B) Solar Energy for Desalination and Drying; (C) Solar Energy for Building Heating and Cooling; and (D) Direct Solar Production of Electricity.

Among these possible uses for solar energy, the heating of buildings and hot water is the most advanced and the most competitive with fuels. Practical systems are available which require the use of glass-covered panels on part of the roof and a small space for overnight heat storage, usually in a basement. An auxiliary furnace supplies heat when necessary - usually about one-fourth the annual demand for space heat and hot water.

There are now upwards of 200 solar heated buildings in the United States. A few of them are well-engineered and instrumented systems, primarily in universities and research institutes, such as the Colorado State University Solar House I, II, and III, the MIT House IV, and etc.. Comparison with the present costs of conventional energy shows that, solar heating is not competitive with natural gas at today's prices, but it is considerably cheaper than electric resistance heating in most parts of the country [3].

Although solar heating systems are now on the market, research and development are still needed to make them better. In this thesis, a research result is reported on Solar Energy for Building Heating and Cooling. Three models of solar heating and cooling systems were simulated on a digital computer using C.S.M.P. (Continuous System Modeling Program). MODEL A, called

SOLAR ENERGY HEATING AND CONVENTIONAL COOLING SYSTEM, is mainly composed of a solar collector, an energy storage tank, an electric hot water heater, an electric furnace to supplement solar energy in the winter, a conventional air conditioner for air-conditioning in the summer, and the necessary pumps, piping, and controls. MODEL B, called SOLAR-ASSISTED HEAT PUMP FOR HEATING AND COOLING, is similar to MODEL A, with the exception that a heat pump is used in place of the air conditioner for house cooling. The heat pump is also used for house heating. MODEL C, called SOLAR-ASSISTED HEAT PUMP FOR HEATING AND HEAT PUMP CHILLED WATER FOR COOLING, is equipped all the same as MODEL B, but the ways of operating the heat pump for house cooling are different.

The simulations were made for a typical two story house located in Manhattan, Kansas. The hour-by-hour performance of each model was studied using the 1976-1977 hourly weather bureau data for the Manhattan area. Several collector areas were tested in these models and a cost analysis was used to determine the minimum cost system. The results also indicate, under what conditions in Manhattan, which of the solar energy systems could be economically competitive with conventional systems.

CHAPTER 2. LITERATURE REVIEW

GENERAL REVIEW

Space heating with solar energy, after about 35 years of limited development effort, has recently become a commercial reality, and it is likely to grow into a major energy supply and industrial activity. The most common type of system for space heating with solar energy involves heat collection in a liquid passing through a flat-plate collector and storage in a tank of hot water.

The trend in liquid collector design and construction is toward a factory-built metal box containing a metal absorber plate coated with black paint or a selective radiation surface, and provided with insulation beneath the metal absorber. Tubes in the absorber plate provide passages for liquid circulation, and one or more transparent covers, usually glass, restrict heat escape. The assembly of individual collector modules into an array of adequate capacity involves piping connections beneath or between adjacent collector panels.

Collectors require several types of materials, the absorber plate being copper, aluminum, or steel, the coating being a dursble black paint or a thin oxide film on a bright metal substrate, and the glazing being a durable, heat resistant plastic or glass, usually tempered.

Throughout most of the United States, freezing may occur on winter nights, so water must not remain in a typical collector when no heating is taking place. One method for dealing with this problem is an automatic draining of the collector whenever the circulating pump is not operating. The other method is by use of a non-freezing liquid in the collector.

Transfer of heat from the storage tank to the building is accomplished either by circulation of heated water through radiators or convectors in

the rooms, or by its circulation through a fan coil exchanger in the duct of a warm air heating system.

Although not as widely utilized as liquid types, a few solar air heating systems have been experimentally used over the past three decades. In its simplest form, a solar air collector closely resembles the liquid heating type. The essential difference is the circulation of air in contact with, and usually beneath, the black absorber plate. Rather than being in tubes, the air may contact the entire surface of the absorber plate. To enhance the heat transfer coefficient, fins, corrugations, or other types of extended surfaces on the absorber may be employed.

Heat storage in an air system, for the best collection efficiency, should also extract substantially all the useful heat from the hot air stream so that the return air can be at the same favorable low temperature as experienced with air returning from the rooms. A highly stratified storage capability is therefore desirable. The thermal properties of loose solids, such as gravel of uniform size, are ideally suited to this application. The very large surface area of the pebbles (commonly 2 to 3 cm in diameter) and the high porosity of a bed of uniformly sized solids results in rapid transfer of heat.

Supply of heat to the rooms from the storage unit is easily accomplished by circulating room air through the pebble-bed in a direction opposite to that employed in the storing cycle. Room air is thus heated by contact with the heated rocks, leaving the pebble-bed in the region of highest rock temperature.

Although long-term performance data on liquid systems for space heating are not available, certain comparisons can be drawn between the liquid and air types. Since the air system is permanently dry, no significant corrosion

of steel or aluminum occurs. There is no possibility of freezing the collector fluid, so decomposable antifreeze compounds or other organic liquids are not required. Boiling can not occur, so replacement of collector fluid is never needed. Fluid leakage, although not precluded, involves minimal corrective cost and no secondary damage. Some long-term operating experience with an air system shows that maintenance is not a significant expense and that the equipment can be expected to have a life comparable to that of the building itself.

Although the basic materials in the air system may be cheaper than those in the liquid type, more space in the building is required for the heat storage units and connecting ducts. Interconnecting conduits and fluid handling equipment may also be somewhat more expensive in air systems.

Löf, et al. [4] compared the performance of solar heating with air and liquid systems and found that under equal conditions, both the liquid and the air solar heating systems can provide space heating (and expected service water heating) with nearly equal performance. They concluded that an air system should have more appeal for residential space heating where low value space can be provided and where maintenance must be low. Liquid solar heating system may be favored in larger commercial space heating installations where maintenance is available and space for heat storage and equipment conduits is of high value.

The addition of solar cooling to heating can evidently improve the economics of the combined process over heating alone. According to the review of Duffie and Beckman [5], three major methods of solar air-conditioning have been investigated. These are: solar sorption cooling; solar-mechanical systems; and solar-related systems which are not solar operated, but which use some components of the solar heating system for cooling.

Operation of absorption air conditioners with energy from flat-plate collector and storage systems is the most common approach to solar cooling today. Heat-operated cooling cycles, based on absorption of refrigerant in liquid absorbent solutions or absorption of refrigerant on solid adsorbents, can be operated by supplying solar energy to accomplish the generation step. These may be closed cycle, e.g. the $LiBr-H_2O$ or NH_3-H_2O cycles, or open cycle in which the refrigerant is water which is exchanged with the atmosphere.

A solar-mechanical cooling system that has received some attention in recent years couples a solar-powered Rankine cylcle engine with a more or less conventional air conditioning system. The problems associated with this system are basically the problems associated with generating mechanical energy from solar energy. The storage tank transfers energy to a heat exchanger which, in turn, transfers energy to a heat engine. The heat engine exchanges energy with the surroundings and produces work.

When solar energy equipment is installed for the purpose of heating a building, some of this equipment can be used to cool the building, but without the direct use of solar energy. These systems are classified as solar-related air conditioning systems, such as heat pump systems, sky radiation systems, rock bed regenerator systems, etc.

Combining the solar system with the heat pump for heating and cooling has recently recieved much attention. The collector, or a storage tank heated by the collector, can serve as the heat pump heat source. Because the heat pump can extract heat from the storage tank at a lower water temperature, collector radiation and convection losses would decrease, resulting in a higher collector efficiencies.

In a solar-heat pump system which supplies heating for a residence, the heat pump can be used to supply cooling during the summer. In this case, cooling is supplied in a conventional manner and solar energy is not utilized. (During the summer solar energy in such a system will probably be used to supply service hot water.) In the heating mode, solar-heat pump systems can be considered in three distinct configurations. In the in-line mode the heat pump is between the storage tank and the load and all energy supplied to the heat pump comes from the solar storage tank. In the parallel mode, solar energy is used to supply heating in the conventional manner and the heat pump, which uses ambient air as the source, acts as the auxiliary for the solar energy system. In the dual source mode, the source of energy for the heat pump can be either the solar heated tank or ambient air depending upon the particular environmental conditions that exist at that time.

It is clear that there is a wide spectrum of potentially interesting methods for using solar energy for building heating and air-conditioning, including many not mentioned in this brief review. Some of these have been extensively studied while some are still completely undeveloped. Many simulation/analysis program have been developed for investigating these systems, such as TRNSYS, SOLAR, NECAP, MINI-SHAC, CERES, and CAP [6]. The first three were developed for simulating solar heating and cooling systems while the latter three were developed to supplement the first three by including rapid long term performance prediction and economic assessment. These programs make the analysis of solar heating and cooling system easier and quicker.

The future of solar heating appears assured by the growing scarcity of fossil fuels and by continued improvements in solar technology. The future

of solar cooling is linked to component (collector and storage) development. It is also closely interrelated with developments in solar heating as the economics of the combined processes appear to be better than that of either one alone.

THE SIMULATION METHOD

There have been two approaches to the design of solar energy systems. The first has been to develop a method which provides an estimate of long-term system performance from suitable averaged meteorological data. The second and more popular approach has been to formulate a computer simulation program of the system and use a computer directly as a study tool.

The transient behavior of solar energy systems using mathematical simulation models has been studied since 1967. The technique is to write an energy balance equation for each component of the system. In general, the equations will be coupled so that it is necessary to find the simultaneous solution. For example, the collector performance is expressed in terms of the temperature of the fluid entering the collector. This in turn, for many systems, is the same as (or is closely related to) the temperature in the exit portion of the storage unit. The outlet temperature from the collector becomes the inlet temperature to the storage unit. In these equations, time is the independent variable and the solution is in the form of temperature as a function of time. In addition, it is possible to integrate energy quantities over time and thus assess the energy distributions for the system.

This approach can be used to estimate, for a particular process application, the amount of energy delivered from the solar collector to meet a load and the quantity of auxiliary energy required. The simulation also indicates

whether the temperature variations for a particular system design are reasonable, for example, whether a collector temperature would rise above the boiling point of the liquid being heated.

The ability to predict dynamic behavior and make energy balances over time is useful in several respects. The analysis provides a means of estimating the effects of design-variable changes on system performance; these design variables might for example, include selectivity of the absorbing surface, number of covers on the collector, collector area, and so on. In this context the system analysis provides a means of understanding how these systems function and can be a guide for experimentation.

Computational simplicity in a simulation model is needed to allow an examination of the long-term performance of many system designs in a variety of climates at reasonable computing cost. Minor details in the component models, from which the system model is formulated, have only a small effect upon the long term system performance and should be excluded. A properly formulated simulation model provides an estimation of the performance of a system more quickly and cheaply than is possible by experiment.

CONCLUSIONS OF OTHERS STUDIES

In this section, some major conclusions which have resulted from other studies of solar energy systems are reviewed. Sheridan, Bullock, and Duffie [7] were among the first to study the transient behavior of solar energy systems using mathematical simulation methods. The only major conclusion of this study is that computers are the best design tool for solar processes.

In a paper pertaining more to solar water-heating than to solar space heating, Close [8] presented a method which he contends is useful in the analysis and design of all transient systems of this type. His approach

begins with the formation of a mathematical model. The model is cast into dimensionless form and the major dimensionless variables of the system are identified. Then factorial analysis, a statistical design technique, is used to determine the effect of these design variables upon the system performance or economics from simulation studies with a digital computer model.

He showed that a representation of storage can be achieved by dividing it into several hypothetical sections which are assumed to be separate but fully mixed. He compared the results for a three-section tank and a six-section tank and concluded that only small differences exist for the two models.

Tybout and Löf [9,10] developed a digital computer simulation program of a solar heating system; they used a model of collector performance and an estimation of the space and water heating requirements to calculate hourly energy balances for a fully-mixed fluid storage tank. The space heating load is estimated by the simple energy per degree-time model. The yearly cost of a solar heating system was estimated by calculating the yearly cost of auxiliary energy required and adding to this cost a yearly payment upon the estimated initial investment.

Despite the simplicity of their model, the economic study of Löf and Tybout [10] has resulted in a number of important conclusions, most of which are in agreement with the conclusions of other. Their conclusions are:

(1) The effects the collector thermal capacitance upon the overall system performance are negligible for all common flat-plate collector constructions.

(2) Either one or two glass cover collectors with non-selective absorber plate surfaces were optimum in even the coldest climate.

(3) The tilt of the collector from horizontal for most economical operation of solar heating systems in winter should be the latitude plus ten to twenty degrees.

(4) An auxiliary energy furnace must be included in a solar space heating to improve the economics as well as to ensure reliability. It must be of the same capacity as that for a conventional heating system.

(5) The optimum energy storage capacity of solar heating systems is the thermal equivalent of ten to fifteen pounds of stored water per square foot of collector area.

Based on 1970 fuel costs and very optimistic collector costs ($2/ft^2), they [10] also concluded that solar heating may be competitive with conventional heating systems; their simulations indicated that the cost of solar heating was somewhat more expensive than heating with natural gas or oil, but less than with electricity in six of the eight locations examined.

The long-term performance of the Colorado Solar House has been reported [11]. This residential solar heated house, which was designed and operated by Löf [11], has been operating continuously since 1957 with no maintenance. The house has 296 m^2 of living area. The collector is an Overlapped-Glass Plate Solar-Air Heating having a cover glass area of 49.1 m^2. Solar heat is stored in a rock bed of primarily granitic rock approximately 1.3 - 2.5 cm in diameter. The results showed that approximately 20% of the total heat load was from solar energy.

Klein, Beckman, and Duffie [12] designed a solar space and water heating system for residences. The system considered uses liquids, either water or antifreeze solution, as the energy transfer and storage medium. A flat-plate

collector is used to transform incident solar radiation into thermal energy. This energy is stored in the form of sensible energy and used as needed to supply the space and water heating load. In their simulation study, the Hottel and Whillier [13] collector equation was used and the hourly diffuse and total radiation on a horizontal surface were estimated using the results of Liu and Jordan [14]. A simple space heating load model, the energy per degree-time model (to be discussed in Chapter 3), was used to estimate the heating demand. The simulation was using TRNSYS [15], a general simulation program for solar energy systems. Based on the collector cost of $\$80/m^2$, they concluded that the fuel costs must be about $\$6/10^6$KJ ($\$6.33/10^6$Btu) before a dollar savings using solar heating can occur.

Hunn, et al. [16] developed a model which simulates the effects of hourly weather conditions on the performance and cost of a combined solar/conventional heating system for building in cold, cloudy climates. The result showed that the solar heating of a typical house in cold, cloudy climates is economically competitive with fuel oil heating only if the price of oil doubles to approximately 80¢/gal.

Extension of the heating study to include summer cooling was made by Löf and Tybout [17]. The mathematical model of their systems was modified to include heat-operated absorption cooling units of the aqueous lithium bromide type. Approximately one hundred analyses of hourly cooling and heating performance for a full year in eight locations were made. They concluded that the use of solar energy for both heating and cooling (including water heating) offers lower costs than does either application alone. Optimum storage for heating and for combined use is about 50 Kg per m^2 of collector area.

Another simulation study of solar heating and cooling system was accomplished by Butz, et al. [18]. Their system includes a water heating collector, a water storage unit, a service hot water facility, a lithium bromide-water air conditioner (with cooling tower), an auxiliary energy source, and associated controls. The simulation was made in the climate of Albuquerque, New Mexico. The results showed that this system would be competitive with conventional means of heating and cooling.

Solar heat pump systems have recently recieved widespread attention. The early analysis by Jordan and Threlkeld [19,20,21] and Löf [22] showed that solar heat pump systems are economically feasible through most of the United States. The General Electric Phase 0 report [23] extended the study of solar-augmented heat pumps to include six types of buildings in nine locations with a variety of different heating and cooling modes. A significant advantage was shown for systems with dual evaporators that switched between ambient and storage to select the highest temperature heat source.

Simulations of the detailed dynamic behavior of solar-augmented heat pump systems have been made by Freeman, et al. [24] and Bosio and Suryanarayana [25]. From these and other studies [26,27], a few basic solar heat pump system configurations have emerged as potentially the most economically advantageous. These systems have the following features in common: an "in-line" heat pump to elevate the temperature of solar energy stored on the evaporator side, the capability to utilize ambient air as the heat pump source when storage is depleted, the capability to bypass the heat pump when storage temperature is high enough, rejection of energy to ambient air in the cooling mode, "second-stage" auxiliary space heating by duct heaters located downstream of the heat pump condenser coil, and a preheater for domestic hot water using "unassisted" solar energy - i.e., collected solar energy which is not augumented by the heat pump.

Many other simulation results for solar heating and cooling systems have been reported in the literature. Most of them have similar conclusions to those mentioned above so have not been reviewed. Based on some of these important conclusions, three solar heating and cooling systems were modeled and simulated in this study. They are discussed in the next chapter.

CHAPTER 3. SIMULATION MODELS

A feasibility study of solar energy heating and cooling systems was made using a digital computer simulation method for the climate of Manhattan, Kansas. Three solar space heating and cooling systems (including hot water service) which use water as the energy collecting and storing medium, were modeled, examined, and compared in this study. In order to make the programming easier and quicker, the simulation programs were written using C.S.M.P. (Continuous System Modeling Program). In writing the programs, the connecting pipes were assumed well-insulated, with no heat loss through them.

A typical two story house located at Manhattan, Kansas was chosen as the simulations basis. The hourly weather bureau data of 1976-1977 were used in the simulations. The three models were: MODEL A - SOLAR ENERGY HEATING AND CONVENTIONAL COOLING SYSTEM; MODEL B - SOLAR-ASSISTED HEAT PUMP FOR HEATING AND COOLING; and MODEL C - SOLAR-ASSISTED HEAT PUMP FOR HEATING AND HEAT PUMP CHILLED WATER FOR COOLING.

MODEL A - SOLAR ENERGY HEATING AND CONVENTIONAL COOLING SYSTEM

A schematic diagram of the MODEL A is shown in Figure 3.1. A solar collector, an energy storage tank, an auxiliary electric furnace, an electric hot water heater and the necessary pumps, fan, and controllers were equipped for house and hot water heating. A conventional air conditioner was used for house cooling in the summer. The simulation was mainly concerned with the weather, the energy collection, the collected energy storage, the loads, and the auxiliary energy. They were discussed separately in the following pages.

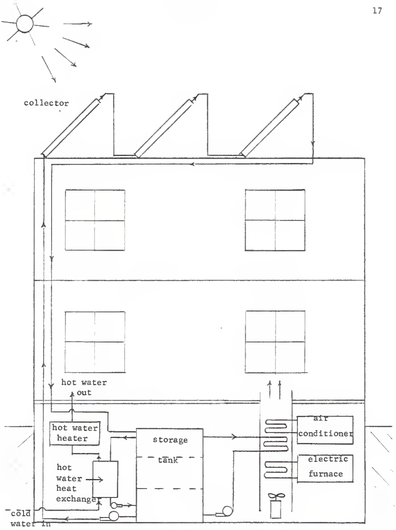

Figure 3.1 Solar Energy Heating and Conventional Cooling System.

THE HOUSE

A well-insulated, 40 x 25 x 18 ft (length x width x height), two story house with main door facing north was selected for the simulations. The house was assumed to have a flat roof and the area of the windows and doors were assumed to be 20% of the total side wall area. The basement was assumed to need no heating or cooling.

ENERGY COLLECTION

The flat-plate collector, the collector pump, and the control system are involved in collecting solar energy and transferring energy to the storage tank. The important parts of a typical flat-plate solar collector, as shown in Figure 3.2, are: the "black" solar energy absorbing surface, with means for transferring the absorbed energy to a fluid; envelopes transparent to solar radiation over the solar absorber surface which reduce convection and radiation losses to the atmosphere; and back insulation to reduce conduction losses as the geometry of the system permits.

The solar collector was modeled based on the information from the book of Duffie and Beckman [28]. As recommended by them, the thermal capacitance of the collector was neglected. The rate of energy collection, Q_u, of a flat-plate collector was expressed as:

$$Q_u = A_c F_R [I_c(\tau\alpha) - U_L(T_{f,i} - T_a)] = (\dot{m}\ c_p)_c\ (T_{f,o} - T_{f,i}) \qquad (1)$$

The collector heat removal factor F_R, which relates the actual useful energy gain of a collector to the useful energy gain if the whole collector surface were at the fluid inlet temperature, was expressed by the equation:

$$F_R = \frac{(\dot{m}\ c_p)_c}{A_c U_L} [1 - \exp(-\frac{A_c U_L F'}{(\dot{m}\ c_p)_c})] \qquad (2)$$

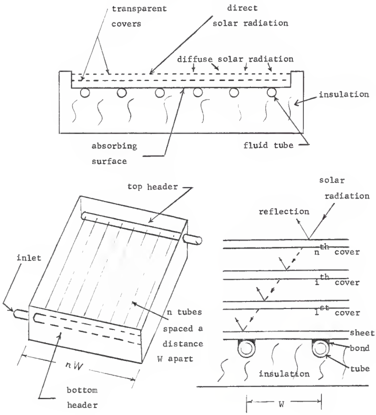

Figure 3.2 The Flat Plate Sheet and Tube Solar Collector.

Where F', the collector efficiency factor, which represents the ratio of the useful energy gain to the useful energy gain if the collector absorbing surface had been at the local fluid temperature, was expressed as:

$$F' = \frac{\frac{1}{U_L}}{W\left[\frac{1}{U_L\,[D+(W-D)F]} + \frac{1}{C_b} + \frac{1}{\pi D_i h_{f,i}}\right]} \qquad (3)$$

The fin efficiency, F, was defined as:

$$F = \frac{[\tanh\, m\ (W-D)/2]}{m\ (W-D)/2} \qquad (4)$$

In the above equation, Eq. (4), m is defined as:

$$m = \left(\frac{U_L}{k\delta'}\right)^{\frac{1}{2}} \qquad (5)$$

In order to apply Eq. (1) to the simulation work, the collector overall energy loss coefficient, U_L, needs to be calculated first. The following equation for U_L was as developed by Klein [29]:

$$U_L = \frac{1}{\dfrac{3.829\ N}{\dfrac{C}{T'_p}\left[\dfrac{T'_p - T'_a}{N + f}\right]^{0.33}} + \dfrac{1}{h_w}} + \frac{\sigma({T'_p}^2 + {T'_a}^2)\ (T'_p + T'_a)}{\dfrac{1}{\varepsilon_p + 0.05N(1-\varepsilon_p)} + \dfrac{2N+f-1}{\varepsilon_g} - N} + U_{be} \qquad (6)$$

The equations for calculating T_p, h_w, C, and f were as follows:

$$T_p = T_{f,i} + \frac{Q_u/A_c}{U_L . F_R}\left[1 - \frac{F_R}{F'}\right] \qquad (7)$$

$$h_w = 1.004 + 0.3\ V \qquad (8)$$

$$C = 365.9\ (1 - 0.00883\varepsilon + 0.0001298\varepsilon^2) \qquad (9)$$

$$f = (1 - 0.227h_w + 0.016h_w^2)\ (1 + 0.091N) \qquad (10)$$

The major assumptions leading to the above simulation equations were:

(1) Performance is steady-state.

(2) Construction is of sheet and tube type.

(3) The headers cover a small area of collector and can be neglected.

(4) The headers provide uniform flow to tubes.

(5) There is no absorption of solar energy by covers insofar as it affects losses from the collector.

(6) There is one dimensional heat flow through covers.

(7) There is a negligible temperature drop through a cover.

(8) There is one-dimensional heat flow through back insulation.

(9) The sky can be considered as a blackbody for long-wavelength radiation at an equivalent sky temperature.

(10) Temperature gradients around tubes can be neglected.

(11) The temperature gradients in the direction of flow and between the tubes can be treated independently.

(12) Properties are independent of temperature.

(13) Loss through front and back are to the same ambient temperature.

(14) Dust and dirt on the collector are negligible.

(15) Shading of the collector absorbing plate is negligible.

(16) The mean plate temperature was equal to the mean fluid temperature in the collector.

The collector was designed to face south and had a tilt angle of 49° measured from horizontal. It was designed to be able to drain water to avoid freezing or boiling. It was "operated" whenever $T_{f,o}$, the collector outlet fluid temperature, was less than 210°F and was greater than $T_{f,i}$, the collector inlet fluid temperature. The major design parameters in the simulation for the collector portion were summed up in Table 3.1.

Table 3.1 Design Parameters of the Solar Collector.

Parameter	Value	Comments
A_c	400;600;800;or 1000 ft^2	several collector areas were simulated
$(\dot{m}c_p)_c$	10 Btu hr^{-1} $°F^{-1}$	per square foot collector area
$(\tau\alpha)$	0.81	assumed independent of incident angle
W	0.328 ft	distance between tubes
D	0.069 ft	outside tube diameter
C_b	∞	neglect bond resistance, see Figure 3.2
D_i	0.049 ft	inside tube diameter
$h_{f,i}$	265 Btu $hr^{-1}ft^{-2}°F^{-1}$	forced circulation
k	121.4 Btu $hr^{-1}ft^{-1}°F^{-1}$	thermal conductivity of Aluminum
N	2	number of glass covers
δ'	1.64×10^{-3} ft	fin thickness
ε_p	0.95	emittance of plate
U_{be}	0.16 Btu $hr^{-1}ft^{-2}°F^{-1}$	back insulation heat loss coefficient

ENERGY STORAGE

Collected solar energy, in the form of heated water, was stored in the storage tank. The stored heated water was supposed to supply all the heat needed for house and hot water heating, but owing to the shortage of winter sunshine and the limitation of storage tank capacity, a supplementary furnace and hot water heater also must be installed.

The storage tank was designed to operate with two circulation loops: the domestic water heating circulation loop and the house heating circulation loop (as shown in Figure 3.3). In the water heating circulation loop, the water from section 1 of the storage tank was circulated through a heat exchanger, in which it exchanged heat with incoming cold water, and then returned to the storage tank section 3. The temperature of the domestic hot water leaving the heat exchanger was assumed 10°F lower than the temperature of water in section 1. When the temperature of the domestic hot water was not as high as the minimum acceptable hot water temperature (140°F), the hot water heater supplied energy to heat it to that temperature.

The hot water needed was assumed to be uniform at 47.3 lb/hr from 7 AM to 7 PM every day. The temperature of the domestic cold water was assumed to be 60°F. When the temperature of water in section 1 of the storage tank was lower than 70°F, the circulation to the heat exchanger was stopped and supplementary heater was used to supply all the hot water heating load. When there is no need of hot water, the circulation is also stopped. When operated, the water circulation rate in this loop was assumed constant value at 1000 lb/hr.

In the house heating circulation loop, the water from section 1 of the storage tank was circulated through coils, heated room air for the house, and then returned to section 3 of the storage tank. This circulation loop

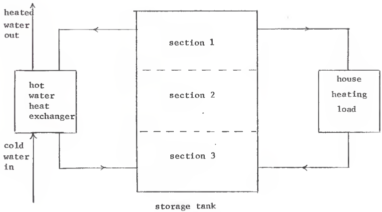

Figure 3.3 The Storage Tank Operation for House Heating and for Hot Water Heating.

was designed to operate at a minimum temperature of 120°F. Once the temperature of water returning from the load was below 120°F, the controller would stop the circulation pump and the supplementary electric furnace would supply all the house heating load. The water circulation rate, when operated, was 7000 lb/hr.

A three-section water storage tank model was used in the simulations. Each section was assumed to have the same capacity and was assumed to be fully mixed. The energy balance equations for the three sections can be written as:

For the top section (section 1)

$$(m\,c_p)_{s,1}\frac{dT_{s,1}}{dt} = F_1(\dot{m}\,c_p)_c(T_{f,o} - T_{s,1}) + F_2(\dot{m}\,c_p)_{HL}(T_{s,2} - T_{s,1}) + F_3(\dot{m}\,c_p)_{LW}(T_{s,2} - T_{s,1}) - (UA)_{s,2}(T_{s,1} - T_B) \tag{11}$$

For the second section (section 2)

$$(m\,c_p)_{s,2}\frac{dT_{s,2}}{dt} = F_1(\dot{m}\,c_p)_c(T_{s,1} - T_{s,2}) + F_2(\dot{m}\,c_p)_{HL}(T_{s,3} - T_{s,2}) + F_3(\dot{m}\,c_p)_{LW}(T_{s,3} - T_{s,2}) - (UA)_{s,2}(T_{s,2} - T_B) \tag{12}$$

For the bottom section (section 3)

$$(m\,c_p)_{s,3}\frac{dT_{s,3}}{dt} = F_1(\dot{m}\,c_p)_c(T_{s,2} - T_{s,3}) + F_2(\dot{m}\,c_p)_{HL}(T_{Lr} - T_{s,3}) + F_3(\dot{m}\,c_p)_{LW}(T_{LW} - T_{s,3}) - (UA)_{s,3}(T_{s,3} - T_B) \tag{13}$$

Table 3.2 Design parameters of the Storage Tank.

Parameter	Value	Comments
$(mc_p)_{s,1}$	4.17 Btu F^{-1}	per square foot of collector area
$(mc_p)_{s,2}$	4.17 Btu F^{-1}	same as above
$(mc_p)_{s,3}$	4.17 Btu F^{-1}	same as above
$(\dot{m}c_p)_{HL}$	7000 Btu $hr^{-1}\,{}^\circ F^{-1}$	a constant
$(\dot{m}c_p)_{LW}$	1000 Btu $hr^{-1}\,{}^\circ F^{-1}$	a constant
$(UA)_{s,1}$	11.6 Btu $hr^{-1}\,{}^\circ F^{-1}$	since the value is small, was assumed a constant for any tank size
$(UA)_{s,2}$	8.49 Btu $hr^{-1}\,{}^\circ F^{-1}$	same as above
$(UA)_{s,3}$	8.49 Btu $hr^{-1}\,{}^\circ F^{-1}$	same as above
T_B	40°F	a constant

In the above equations, T_{Lr} and T_{LW} will be discussed in the following sections. Values of F_1, F_2, and F_3 are determined by:

$$F_1 = \begin{cases} 1 & \text{when } T_{f,o} > T_{f,i} \text{ and } T_{f,o} < 210°F \\ 0 & \text{otherwise} \end{cases} \tag{14}$$

$$F_2 = \begin{cases} 1 & \text{when } T_{Lr} > 120°F \\ 0 & \text{otherwise} \end{cases} \tag{15}$$

$$F_3 = \begin{cases} 1 & \text{When hot water is needed and } T_{s,1} > 70°F \\ 0 & \text{otherwise} \end{cases} \tag{16}$$

The tank was assumed to be a light gauge, vertical, well-insulated, galvanized, steel cylinder located in the basement. The size of the thermal storage tank is primarily an economic decision. Based on the economic studies of Tybout and Löf [10], the thermal equivalent of 12.5 pounds of stored water per square foot of collector area was adopted. Other major design parameters are summed up in Table 3.2.

DOMESTIC HOT WATER LOAD

The hot water heating load, Q_{WL}, was assumed constant at 3784 Btu/hr every day from 7 AM to 7 PM*. Since the heated water leaving the heat exchanger was assumed 10°F lower than the temperature of water in section 1 of the storage tank, the amount of hot water heating load supplied by the solar energy system can be expressed as:

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* $$Q_{WL} = \begin{cases} (47.3\ \frac{lb}{hr})(1\ \frac{Btu}{lb\ °F})(140-60)(°F) = 3784\ \frac{Btu}{hr} & \text{(7 AM to 7 PM)} \\ 0 & \text{otherwise} \end{cases} \tag{16A}$$

$$Q_{BS} = (47.3)(1)(T_{hot} - 60) \tag{17}$$

where:

$$T_{hot} = T_{s,1} - 10 \tag{18}$$

And, the temperature of water returning from the hot water heat exchanger to section 3 of the storage tank was calculated by:

$$T_{LW} = T_{s,1} - \frac{Q_{BS}}{(\dot{m}c_p)_{LW}} \tag{19}$$

HOUSE HEATING LOAD

The house heating load for a building is just equal to the heat losses from the building. The heat losses may be divided into two groups: (1) the transmission losses, or heat transmitted through the confining walls, floor, ceiling, glass or other surfaces; and (2) the infiltration losses, or heat required to warm outdoor air which leaks in through cracks and crevices, around doors and windows, or heat required to warm outdoor air used for ventilation.

In this study, a simple space heating model, the energy per degree-time model [12], was used in estimating the heating load. The house heating load Q_{HL} was expressed as a linear function of the difference between the inside temperature of the building and the ambient temperature,

$$Q_{HL} = (U)(A)(T_i - T_a) \tag{20}$$

The design indoor temperature was 70°F. The value of UA for the simulation house was calculated based on ASHRAE estimation methods [30] as in Table 3.3 and was equal to 1014 Btu $hr^{-1}{}^{\circ}F^{-1}$.

The assumptions made in calculating UA were as follows:

(1) The residence was assumed to be heated as a single zone and was assumed to be occupied, always heated, for 24 hours per day, every day of the heating season.

Table 3.3 Calculation of the House Total Equivalent Heat Loss Coefficient (UA).

ITEM	COMPONENT	AREA (ft^2)	CONSTRUCTION	RESISTANCE ($Btu^{-1} hr\ ft^2 {}^\circ F$)	U ($Btu\ hr^{-1} ft^{-2} {}^\circ F^{-1}$)	UA ($Btu\ hr^{-1} {}^\circ F^{-1}$)
TRANSMISSION HEAT LOSS	Wall	1872	outside surface	0.17	0.077	211
			0.5" plaster board	0.357		
			insulation 3.5"	11		
			wood siding 1"	0.781		
			inside surface	0.68		
	Roof	1000	outside surface	0.17	0.17	170
			built-in roofing	0.33		
			roof insulation	1.39		
			plywood deck	0.78		
			air space	0.85		
			gypsum wallboard	0.45		
			acoustical tile	1.25		
			inside surface	0.61		
	Window & Door	468	double window (0.5" air space, emisivity 0.4)		0.45	144
INFILTRATION HEAT LOSS	Sensible Heat Loss		No. of air change per hour = 1.5			489
House Total Equivalent Heat Loss Coefficient, UA =						1014

(2) The basement was assumed to need no heating or cooling.

(3) The transient heat from the storage tank warms the air near the basement ceiling sufficiently to make it unnecessary to make an allowance for floor heat loss from rooms located over the basement.

(4) The humidity difference between the indoor air and the outdoor air was negligible.

(5) The heat supplied by persons, lights, motors, machinery, and sunshine were negligible.

(6) The wind speed was assumed to be a constant, 15 M.P.H.

Once the heating load was calculated by equation (20), the determination of T_{Lr} was made by the following equation:

$$T_{Lr} = T_{s,1} - \frac{Q_{HL}}{(\dot{m}c_p)_{HL}} \tag{21}$$

HOUSE COOLING LOAD

The loads on residential cooling systems are primarily those imposed by heat flow through structural components and by air leakage or ventilation. In this study, the calculation of cooling load was based on the estimation methods of ASHRAE [30] and Allen [31]. The residence was assumed to be cooled as a single zone, and usually conditioned, for 24 hours per day, every day of the cooling season.

The total residential cooling load is the sum of the sensible heat load and the latent heat load. A sensible heat load is considered to occur when there is a direct addition of heat to the enclosure by any one or all of the mechanisms of conduction, convection, and radiation. A latent head load is considered to occur when there is an addition of water vapor to the

air in an enclosure. According to ASHRAE [30]; because of the nature of the structure, the occupancy, and the equipment; the latent portion of a residential cooling load is usually estimated as equaling thirty percent of the calculated sensible heat load. The calculation of residential cooling load is therefore invariably the calculation of the sensible heat load.

In calculation of the sensible heat load, the following items must be considered:

(1) heat gain through walls and roof

(2) heat gain through windows

(3) heat gain due to infiltration and ventilation

(4) heat gain due to occupancy loads

The detailed sensible heat load calculations of the above four items are as follows:

(1) heat gain through walls and roof

The heat gain through a wall may be expressed by the equations proposed by Faust, Levine, and Urban [32],

$$H_{wall} = H_{t,wall} + H_{r,wall} \tag{22}$$

and

$$H_{t,wall} = (A_{wall})(U_{wall})(T_a - T_i) \tag{23}$$

$$H_{r,wall} = (A_{wall})(P)(a)(I_a) \tag{24}$$

The values of P, the portion of the absorbed solar radiation that is transmitted to the inside, are related to the transmission coefficient of the wall, and the relationship is approximately

$$P = (0.23)(U_{wall}) \tag{25}$$

Table 3.4 Heat Gain through Walls and Roof.

COMPONENT	AREA (ft^2)	U (Btu $hr^{-1} ft^{-2} {}^\circ F^{-1}$)	TIME LAG (hr)	HEAT GAIN (Btu hr^{-1})
east wall	576	0.077	4	$(44.35)(T_a-75) + (4.08)(I_a)$
west wall	576	0.077	4	$(44.35)(T_a-75) + (4.08)(I_a)$
south wall	360	0.077	4	$(27.72)(T_a-75) + (2.55)(I_a)$
north wall	360	0.077	4	$(27.72)(T_a-75) + (2.55)(I_a)$
roof	1000	0.17	4	$(170)(T_a-75)$

and Eq. (24) becomes

$$H_{r,wall} = (0.23)(A_{wall})(U_{wall})(a)(I_a) \tag{26}$$

The absorption coefficient, a, was assumed to be 0.4 for the light colored wall surfaces. Values of I_a, the solar radiation intensity, vary with time and the direction the walls face.

The calculation of heat gain through the roof is almost the same as the calculation of heat gain through a wall except that it was assumed that the collector was located on the roof, and thus shaded the roof, eliminating heat gain by solar radiation. The equation for calculating the heat gain through the roof was:

$$H_{roof} = (A_{roof})(U_{roof})(T_a - T_i) \tag{27}$$

According to Allen [31], the thermal capacity of the outer walls and roof has a marked effect by causing a time lag in the penetration of heat to the room. This lag may be as much as 3 hour for a 6-in. concrete wall and 10 hour for a 22-in. brick and tile wall. So that, in applying the above two equations, Eq. (22) and Eq. (27), a time lag must be considered. It was assumed to be 4 hours. The heat gain through walls and roof were calculated as in Table 3.4.

(2) heat gain through windows

The heat gain through windows may be expressed by a simple equation:

$$H_w = A_w F_w I_a + A_w U_w (T_a - T_i) \tag{28}$$

The solar heat gain coefficient F_w is a characteristic of each type of fenestration and varies with the incident, since transmittance and absorptance of the glazing material are dependent upon the incident angle. But in this study, it was assumed that transmittance and absorptance of the glazing material were constant, so that, the value of F_w was also assumed independent of the incident angle.

Table 3.5 Heat Gain through Windows.

COMPONENT	AREA (ft^2)	U ($Btu\ hr^{-1} ft^{-2} {}^{\circ}F^{-1}$)	F_w	HEAT GAIN ($Btu\ hr^{-1}$)
east side windows	144	0.44	0.22	$(63.4)(T_a-75) + (31.68)(I_a)$
west side windows	144	0.44	0.22	$(63.4)(T_a-75) + (31.68)(I_a)$
north side windows	90	0.44	0.22	$(39.6)(T_a-75) + (19.8)(I_a)$
south side windows	90	0.44	0.22	$(39.6)(T_a-75) + (19.8)(I_a)$

The calculation of F_w was based on the Shading Coefficient described in ASHRAE [30]. The Shading Coefficient, S.C., is represented by the equation,

$$S.C. = \frac{\text{Solar Heat Gain of Fenestration}}{\text{Solar Heat Gain of Double-Strength Glass}} \tag{29}$$

In terms of the Solar Heat Gain Coefficient, F_w, the Shading Coefficient is a ratio of F_w for the fenestration and F_w for D.S.A. glass which, for standard summer conditions, is 0.87

$$S.C. = \frac{F_w \text{ of Fenestration}}{F_w \text{ of D.S.A.} = 0.87} \tag{30}$$

For insulating glass with indoor shading by opaque, white roller shades, the Shading Coefficient was determined to be 0.25. Based on this value, the Solar Heat Gain Coefficient F_w of the window was calculated by Eq. (30) to be 0.22. The Heat Gain Through Windows was calculated as in Table 3.5.

(3) heat gain due to infiltration and ventilation

Natural air leakage in residential structures is much smaller in summer than in winter. According to ASHRAE [30], Infiltration and Ventilation air flow is usually assumed at a rate of about one air change per hour in summer.

The calculation of heat gain due to infiltration and ventilation was based on the equation (reference (29)):

$$H_I = (0.018)(V_o)(C_a)(T_a - T_i) \tag{31}$$

Since V_o = 18000 ft^3, C_a = 1, and T_i = 75°F, the heat gain due to infiltration and ventilation was calculated as $H_I = (324)(T_a - 75)$.

(4) heat gain due to occupancy loads

Even though occupant density is usually lower in residences than in many other types of structures, occupancy loads must be considered in residential cooling load calculations. Loads are generated both by occupants

themselves, and by household appliances. Such loads must usually be treated in an approximate manner, since occupancy and occupant activities are varied and unpredictable.

According to ASHRAE [30], heat release per occupant of residence is usually assumed to be 225 Btu per hour of sensible heat. Appliance load can be limited to the kitchen in most cases and is assumed to be 1200 Btu per hour of sensible heat. In the simulation model, the number of occupants was estimated to be 6 and the appliance loads were assumed to be at 5-6 AM, 11-12 AM, and 5-6 PM.

The total cooling load is the sum of the sensible load and the latent load. Inasmuch as latent load does not enter directly into residential load calculation, total load is usually calculated as 1.3 times the calculated sensible load.

WEATHER DATA

The weather data needed in the simulation study were hourly solar radiation data, hourly ambient temperature, and hourly wind speed data (for calculating collector heat loss coefficient). These weather data were provided by the Department of Physics, Kansas State University.

The hourly solar radiation data provided were the direct and diffuse solar radiation in gm-cal per cm^2 of horizontal surface. The actual hourly solar radiation impinging on the tilted solar collector, the vertical walls and windows were calculated by first determining the solar angle for every surface at each hour.

According to ASHRAE [30], the incident angle θ is related to the solar attitude β, the wall-solar azimuth γ, and the tilt angle ε, by:

$$\cos\theta = \cos\beta \cos\gamma \sin\varepsilon + \sin\beta \cos\varepsilon \tag{32}$$

Other equations needed for the calculation of $\cos\theta$ are:

$$\sin\beta = \cos L \cos\delta \cos H + \sin L \sin\delta \tag{33}$$

$$\sin\phi = \cos\delta \sin H / \cos\beta \tag{34}$$

$$H = 0.25 \times (\text{Number of minutes from local solar noon}) \tag{35}$$

The solar declination δ, which is a function of the date (see Table 3.6), was assumed to be a constant for each month. The relationship among the above angles is as shown in Figure 3.4.

According to Eq. (30), when a surface is horizontal, ε=0°, then

$$\cos\theta_H = \sin\beta \tag{36}$$

And, for a vertical surface, ε=90°

$$\cos\theta_V = \cos\beta \cos\gamma \tag{37}$$

Also, for the collector facing south (γ=0°), and with a tilted angle of 49° (ε=49°),

$$\cos\theta_{49°} = \cos\beta \sin 49° + \sin\beta \cos 49° \tag{38}$$

Then, the solar radiation impinging on the collector can be calculated by the following equation:

$$I_c = \frac{I_M}{\sin\beta} \times (\cos\theta_{49°}) \tag{39}$$

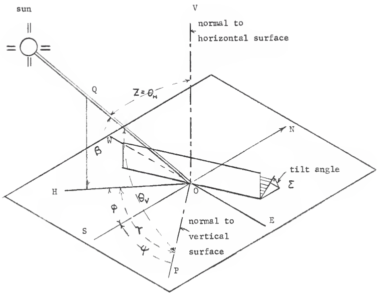
And, for vertical walls and windows facing different directions, the actual solar radiation I_a can be calculated by the equation:

$$I_a = \frac{I_M}{\sin\beta} \times (\cos\beta \cos\gamma) \tag{40}$$

The value of γ, the wall-solar azimuth, in the above equation for differently facing vertical walls and windows are given in Table 3.7.

Table 3.6 Values of Solar Declination.

DATE	DECLINATION , degrees
Jan 21	-20
Feb 21	-10.8
Mar 21	0
Apr 21	+11.6
May 21	+21.0
Jun 21	+23.45
Jul 21	+20.6
Aug 21	+12.3
Sep 21	0
Oct 21	-10.5
Nov 21	-19.8
Dec 21	-23.45



SOLAR ALTITUDE, $\beta = \angle$ QOH

SOLAR AZIMUTH, $\phi = \angle$ SOH

ZENITH ANGLE , $Z = \angle$ QOV

INCIDENT ANGLE, $\theta_v = \angle$ QOP

WALL AZIMUTH, $\psi = \angle$ SOP

WALL-SOLAR AZIMUTH, $\gamma = \angle$ HOP

Figure 3.4 Definition of Solar Angles.

Table 3.7 Values of Wall-Solar Azimuth for Different Facing Vertical Surface

DIRECTION WALL OR WINDOW FACE	γ, degrees	NOTES
east	ϕ - 90 AM ϕ + 90 PM	1. ϕ is calculated by Eq.(34). 2. Treat negative values of
west	ϕ + 90 AM ϕ - 90 PM	γ as if they were positive.
south	ϕ	3. If γ is greater than $90°$, the surface is in the shade.
north	greater than $90°$	

The derivations of Eq. (39) and Eq. (40) were based on the assumption that most of the diffuse radiation comes from an apparent origin near the sun, that is, the scattering of solar radiation is mostly forward scattering. Under this assumption, the horizontal radiation was treated as though it was all beam radiation. Although this approximation is best suitable on clear days, it should not cause large errors in the simulations.

AUXILIARY ENERGY

In this model, the auxiliary energy for house heating and for hot water heating was supplied separately by an electric furnace and an electric hot water heater. The furnace was assumed to be able to supply all the heating load without having any time lag once the controller stops the circulation of heated water from the storage tank to the load (when $T_{Lr} < 120°F$). A similar assumption was made on the hot water heater - i.e., that it could supply energy to supplement the solar energy to meet the hot water load at all times.

The hourly instantaneous auxiliary energy required for house heating and for hot water heating was estimated in the simulation by the following equations:

$$Q_{AUX} = \begin{cases} Q_{HL} & \text{when } T_{Lr} < 120°F \\ 0 & \text{otherwise} \end{cases} \tag{41}$$

$$Q_{AUW} = \begin{cases} 3784 & \text{when } T_{s,1} < 70°F \text{ and hot water is needed.} \\ (47.3)(140-T_{hot}) & \text{when } 60°F < T_{hot} < 140°F \text{ and hot water is needed} \\ 0 & \text{when } T_{hot} > 140°F \text{ or hot water is not needed} \end{cases} \tag{42}$$

And, the long-term cumulative auxiliary energy required for both items was calculated by:

$$\overline{Q}_{AUX} = \Sigma(Q_{AUX})(\Delta t) \tag{43}$$

$$\overline{Q}_{AUW} = \Sigma(Q_{AUW})(\Delta t) \tag{44}$$

where Δt is the time interval; one hour was chosen in the simulations.

In the calculations above, the electric energy consumed by the circulation pumps was neglected. The same assumption was also made in MODEL B and MODEL C.

In the cooling season, the electric energy consumed by the compressor of the air conditioner was estimated by the value of C.O.P. (Coefficient of Performance). This will be discussed in the section - HEAT PUMP. (It was assumed in this study that the C.O.P. of an air conditioner and a heat pump was the same when they are operated in the same weather conditions.)

MODEL B - SOLAR-ASSISTED HEAT PUMP FOR HEATING AND COOLING

A schematic diagram of the heating portion of this model is shown in Figure 3.5. Unlike in MODEL A, a heat pump was installed for both heating and cooling the house. A solar collector, an energy storage tank, an electric hot water heater, an auxiliary electric furnace, a heat pump, and the necessary pumps, fan, and controllers were used for house heating. The heat pump was also used for house air-conditioning in the cooling season. It was assumed to be just like an air conditioner when used for cooling.

Except for the heat pump, the energy storage, and the auxiliary energy, all component simulations were the same for this model as was the case for the previous model. The energy storage, the heat pump and the auxiliary energy were modeled as follows.

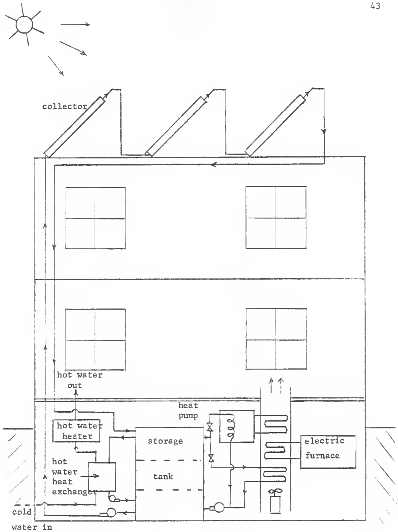


Figure 3.5 Solar-Assisted Heat Pump Heating System.

ENERGY STORAGE

In this model, the water in the storage tank was again designed to operate in two major circulation loops: the hot water heating loop and the house heating loop. The hot water heating loop was modeled the same as in the previous model, and will not be discussed again. In the house heating circulation loop, there are two routes for water circulation. The first route is: water from the storage tank section 1 was circulated through heating coils in which it exchanged heat with room air for house heating and then returned to the storage tank section 3. The second route is: water from the storage tank section 1 was circulated through the heat pump suction side heat exchanger in which it heated the refrigerant and then returned to the storage tank section 3.

When the temperature of water returning from the load (heating coils), T_{Lr}, is higher than 120°F, the circulation pump will circulate the water through the first route, and the stored heated water is responsible for all the house heating load. When T_{Lr} is lower than 120°F and the temperature of water returning from the heat pump heat exchanger, T_{Lh}, is higher than 40°F, the circulation pump will circulate the water through the second route. In this case, the stored heated water is used as a heat source for the heat pump, and the heat pump is responsible for all or part of the house heating load. When T_{Lr} < 120°F and T_{Lh} < 40°F, the controller will stop the circulation pump, and the electric furnace will supply all the heat for house heating.

Such operation can be expressed by previously derived equations, Eq. (11) (12)(13)(14) and (16), except that Eq. (15) is replaced by:

$$F_2 = \begin{cases} 1 & \text{when } T_{Lr} > 120°F \\ 1 & \text{when } T_{Lr} < 120°F \text{ and } T_{Lh} > 40°F \text{ (with } T_{Lr} \\ & \text{replaced by } T_{Lh} \text{ in Eq. (13))} \\ 0 & \text{when } T_{Lr} < 120°F \text{ and } T_{Lh} < 40°F \end{cases} \tag{45}$$

The calculation of T_{Lr} was by Eq. (21), and the calculation of T_{Lh} will be discussed in the next section. Other portions were modeled the same as in MODEL A.

HEAT PUMP

The heat pump was operated in the heating season when $T_{Lr} < 120°F$ and $T_{Lh} > 40°F$, and was operated in the cooling season just like a conventional air conditioner. A heat pump heating cycle is shown in Figure 3.6. The heat pump heat source was the heated water in the storage tank.

The heat pump capacity was assumed to be a function of the heat pump C.O.P. (Coefficient of Performance). The ideal heat pump C.O.P. is defined as:

$$C.O.P._I = \frac{\text{heat rejected}}{\text{work required}} = \frac{T_2}{T_2 - T_1} \tag{46}$$

According to Kemler and Oglesby [33], the ratio of the C.O.P. of a commercial compressor to the ideal C.O.P. is just about one half. This ratio is a weak function of both the suction and the condensing temperature. In this heat pump model, a constant of 0.53 was assumed to be the ratio of a commercial heat pump C.O.P. compared to the ideal C.O.P. Then, the heat pump C.O.P. was calculated by:

$$C.O.P. = \frac{\text{heat rejected}}{\text{work required}} = (0.53)\left(\frac{T_2}{T_2 - T_1}\right) \tag{47}$$

The amount of heat rejected by the heat pump is, from the thermodynamics first law, the sum of the heat absorbed plus the net work input (work required), that is:

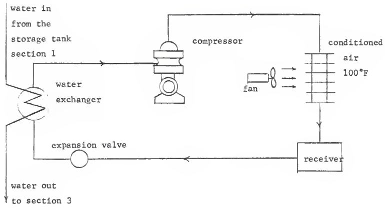


Figure 3.6 The Heat Pump Heating Cycle.

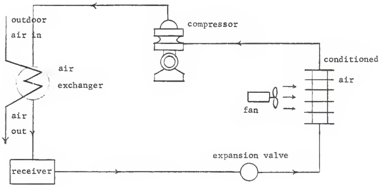


Figure 3.7 The Heat Pump Cooling Cycle.

heat rejected = heat abosrbed + net work input (48)

It is clear from the above equation that the work done by the compressor finally becomes the output heat of the heat pump. By assuming that all the electric energy consumed can be converted through the heat pump compressor into work, the auxiliary energy required (electric energy consumed by the compressor) is thus equivalent to the heat added by the compressor (or work input).

The amount of heat rejected by the heat pump is actually the heat pump capacity. It was determined by rearranging Eq. (47),

$$\begin{aligned}(\text{the heat pump capacity, } C_h) &= (\text{C.O.P.})(\text{work required}) \\ &= (\text{C.O.P.})(\text{horsepower input}) \qquad (49)\end{aligned}$$

The horsepower input was treated as a constant which depended on the size of the heat pump compressor. Once the compressor size was fixed, the heat pump capacity was therefore a function of C.O.P. only. In this heat pump model, the suction temperature T_1 was assumed to be 10°F below that of the temperature of water in the storage tank section 1. The condensing temperature was designed to be 120°F.

When the heat pump was operated for house heating, the procedures of determining T_{Lh}, temperature of water returning from the heat pump, and Q_{AUX}, instantaneous hourly auxiliary energy required, are as follows:

(1) Calculate C.O.P. by Eq. (47)

(2) Determine C_h by Eq. (49)

(3) If $C_h > Q_{HL}$, this means that the operation of the heat pump alone can meet the house heating load, and

$$Q_{AUX} = \frac{Q_{HL}}{\text{C.O.P.}} \qquad (50)$$

$$T_{Lh} = T_{s,1} - \frac{Q_{HL}}{(\dot{m}c_p)_{HL}} \left(1 - \frac{1}{C.O.P.}\right) \tag{51}$$

(4) If $C_h < Q_{HL}$, the electric furnace is operated to supplement the heat pump system, and

$$Q_{AUX} = \frac{C_h}{C.O.P.} + (Q_{HL} - C_h) \tag{52}$$

$$T_{Lh} = T_{s,1} - \frac{C_h}{(\dot{m}c_p)_{HL}} \left(1 - \frac{1}{C.O.P.}\right) \tag{53}$$

When the heat pump was operated for house cooling, it was assumed to operate in the same manner as a conventional air conditioner. A heat pump cooling cycle is shown in Figure 3.7. In this case, the condensing temperature was assumed 20°F above the ambient temperature, and the suction temperature was designed to be 55°F. The value of the C.O.P. was calculated by Eq. (47), and the hourly auxiliary energy required (electric energy consumed by the compressor) was calculated by:

$$Q_{AUX} = \begin{cases} \dfrac{\text{house cooling load}}{\text{C.O.P.} - 1} & \text{when cooling is required (cooling load > 0)} \\ 0 & \text{otherwise} \end{cases} \tag{54}$$

(This assumed that the size of the heat pump or the air conditioner was large enough to meet all the house cooling load.)

AUXILIARY ENERGY

There are three kinds of auxiliary energy required for house and hot water heating in this simulation model:

(1) Electric energy required to operate the heat pump compressor.

(2) Electric energy required to operate the electric furnace.

(3) Electric energy required to operate the hot water heater.

The methods of determining the hourly and cumulative auxiliary energy required for house heating were as follows:

(1) When $T_{Lr} > 120°F$, all the house heating load was supplied by the stored heated water, and

$$Q_{AUX} = 0 \tag{55}$$

(2) When $T_{Lr} < 120°F$, and $T_{Lh} > 40°F$, Q_{AUX} is expressed as in Eq. (50) and (52).

(3) When $T_{Lr} < 120°F$ and $T_{Lh} < 40°F$,

$$Q_{AUX} = Q_{HL} \tag{56}$$

(4) The cumulative auxiliary energy required for house heating was calculated by adding all the hourly auxiliary energy requirement terms during the simulation,

$$\bar{Q}_{AUX} = \Sigma(Q_{AUX})(\Delta t) \tag{57}$$

where Δt is chosen to be one hour.

The hourly auxiliary energy required for house cooling and for hot water heating was calculated by Eq. (54) and by Eq. (42). The long-term cumulative auxiliary energy required for these two items was again calculated by adding all the hourly terms during the simulation.

MODEL C - SOLAR-ASSISTED HEAT PUMP FOR HEATING AND HEAT PUMP CHILLED WATER FOR COOLING

The major components of this model were all the same as in the previous model. The only difference between these two models was the method of operation in the summer. Instead of being operated like an air conditioner in the cooling season, the heat pump in this model was designed to operate to chill the water in the storage tank during the summer night, and this chilled water was used to cool the house.

The heat pump was designed to start operating at midnight every night until the temperature of water in the storage tank section 1 reached 40°F. The chilled water in the storage tank section 3 was circulated through cooling coils to meet the cooling load once the house cooling was needed, and then returned to the storage tank section 1. When the temperature of water in the storage tank section 3 rose to 55°F, the heat pump was called to chill the water so that the temperature of water for cooling would not be above the designed maximum cooling temperature of 55°F.

A schematic diagram of this cooling operation is shown in Figure 3.8. The energy balance equations for the storage tank were as follows:

For the top section (section 1)

$$(mc_p)_{s,1}\frac{dT_{s,1}}{dt} = F_5(\dot{m}c_p)_{CL}(T_{CL} - T_{s,1}) + F_6(\dot{m}c_p)_{CH}(T_{s,2} - T_{s,1}) + (UA)_{s,1}(T_B - T_{s,1}) \tag{58}$$

For the second section (section 2)

$$(mc_p)_{s,2}\frac{dT_{s,2}}{dt} = F_5(\dot{m}c_p)_{CL}(T_{s,1} - T_{s,2}) + F_6(\dot{m}c_p)_{CH}(T_{s,3} - T_{s,2}) + (UA)_{s,2}(T_B - T_{s,2}) \tag{59}$$

For the bottom section (section 3)

$$(mc_p)_{s,3}\frac{dT_{s,3}}{dt} = F_5(\dot{m}c_p)_{CL}(T_{s,2} - T_{s,3}) + F_6(\dot{m}c_p)_{CH}(T_{CH} - T_{s,3}) + (UA)_{s,3}(T_B - T_{s,3}) \tag{60}$$

In the above equations, F_5 and F_6 were expressed as:

$$F_5 = \begin{cases} 1 & \text{when house cooling was needed} \\ 0 & \text{otherwise} \end{cases} \tag{61}$$

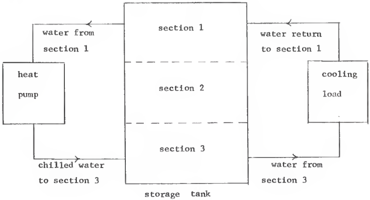


Figure 3.8 Combined the Storage Tank and the Heat Pump for Cooling Operation.

$$F_6 = \begin{cases} 1 & \text{when the heat pump was operated} \\ 0 & \text{otherwise} \end{cases} \tag{62}$$

The temperature of water returning from the cooling coils, T_{CL}, when house cooling is needed, was determined by:

$$T_{CL} = T_{s,3} + \frac{\text{house cooling load}}{(\dot{m}c_p)_{CL}} \tag{63}$$

And, the temperature of water returning from the heat pump, T_{CH}, was calculated by:

$$T_{CH} = T_{s,1} - \frac{C_h}{(\dot{m}c_p)_{CH}} \left(1 - \frac{1}{C.O.P.}\right) \tag{64}$$

In the equation above, the value of C_h and the value of C.O.P. were determined by Eq. (49) and by Eq. (47).

Finally, the hourly auxiliary energy required (electric energy consumed by the heat pump compressor) was calculated by:

$$Q_{AUX} = \begin{cases} \dfrac{C_h}{C.O.P.} & \text{when the heat pump was operated} \\ 0 & \text{otherwise} \end{cases} \tag{65}$$

Again, the cumulative auxiliary energy required is the sum of the hourly terms over all the simulation hours.

In calculating the above equations, the heat pump condensing temperature was assumed to be 20°F above the ambient temperature and the suction temperature was assumed to be 10°F below the temperature of water in the storage tank section 1. The basement temperature was assumed to be 70°F. The values of $(\dot{m}c_p)_{CL}$ and $(\dot{m}c_p)_{CH}$ were designed as 7000 and 12000 Btu/hr°F.

The simulation results of the three models are reported in CHAPTER 5.

CHAPTER 4. CONTINUOUS SYSTEM MODELING PROGRAM

During the last few years the digital computer has assumed a prominent role in system simulation and in the solution of nontrivial mathematical equations. Computer languages such as FORTRAN, BASIC, and PL/1 have provided users with a convenient method for communicating with the computer on a wide range of problems. In many cases, specialized computer programs have been developed that are directly tailored to a particular class of problems. CONTINUOUS SYSTEM MODELING PROGRAM (CSMP) is one of those kind of programs. It is most suitable for the simulation of physical systems.

The CONTINUOUS SYSTEM MODELING PROGRAM (developed by the IBM Company) was designed to allow users to simulate all types of physical systems with a minimum of programming difficulty. It is an application-oriented program in that it is specially written for scientists, engineers, and analysts who are involved in work that requires the solution of ordinary differential equations or in simulating a system that has been modeled as a block diagram.

In developing the program, emphasis was placed on simplified input data statements, output statements and on program control statements that almost directly describe the mathematical equations or physical variables of the problem. The program which is an extension of FORTRAN language includes a basic set of functional blocks which can represent the components of a continuous system and accepts application-oriented statements defining the connections between these functional blocks. Included in the basic set are such conventional analog computer components as integrators and relays plus many special purpose functions like delay time, zero-order hold, dead space, limiter functions, and so on .... Those functions really make the CSMP an

attractive language in simulating complex physical systems. For example, for simulations requiring integration, the user seldom needs to become involved in specifying either the type or any of the details of the numerical integrating method. This simplifies the programming and reduces the chances of making mistakes. In effect, CSMP allows the user to concentrate on the details of the physical system rather than the usual concerns of numerical analysis and programming.

Using the CSMP program is similar in many respects to using the electronic analog computer. Many of the special CSMP statements perform the same function as typical analog computer components. In fact, CSMP digital simulation has many advantages over the use of an analog machine in that (1) is no need to be concerned with amplitude and time scaling; (2) is more accurate; (3) is easier to program; and (4) has much greater capability in handling nonlinear and time-variant problems.

In addition to simulating solar processes, applications in which CSMP can be used include studies of nuclear reactions, control system design, parameter estimation, studies of blood circulation and other physiological processes, studies of chemical refineries, natural gas transmission, process control, investigation of aircraft landing and take-off, plant growth, natural resources management and other industrial dynamics. Anyone working in a field of science who has a need to simulate a system or solve ordinary differential equations will quickly recognize the power of CSMP.

CSMP is most useful for small and medium-size simulations. For extremely large and complex problems, the simulation may be limited by program-size restrictions. Also, because of the flexibility in programming, CSMP simulations may require slightly more computer time than custom-developed programs.

The detailed illustrations of using the CSMP are presented as in References [34,35,36].

CHAPTER 5. SIMULATION RESULTS

HEATING RESULTS

The heating results of interest in this study are long-term integrated energy quantities. These include the cumulative hot water heating load, $\bar{Q}_{WL}$; the cumulative house heating load, $\bar{Q}_{HL}$; the cumulative auxiliary energy added by the heat pump and/or the electric furnace, $\bar{Q}_{AUX}$; and the cumulative auxiliary energy added by the electric hot water heater, $\bar{Q}_{AUW}$.

Two simulation programs were written using CSMP (Continuous System Modeling Program) to calculate the above mentioned terms. The programs were written according to the component models described in Chapter 3. The equations among the component models were coupled and were well arranged so that they gave the correct simultaneous solution. The simulation algorithms were as follows:

(1) Give the initial conditions, such as initial tank temperature, initial plate temperature, etc.

(2) Read the weather data.

(3) Calculate the solar radiation impinging on the collector, I_c, by Eq. (39).

(4) Calculate the collector heat loss coefficient, U_L, by Eq. (6).

(5) Determine the collector outlet fluid temperature, $T_{f,o}$, by Eq. (1). (Assuming that the collector inlet fluid temperature is the same as the section 3 fluid temperature of the storage tank.)

(6) Calculate the hourly hot water heating load, Q_{WL}, and the hourly house heating load, Q_{HL}, by Eq. (16A) and by Eq. (20).

(7) Calculate the cumulative quantities of the above two terms, $\bar{Q}_{WL}$ and $\bar{Q}_{HL}$.

(8) Calculate the temperature of water returning from the hot water heat exchanger, T_{LW}, by Eq. (19).

(9) Calculate the temperature of water returning from the house heating load, T_{Lr}, by Eq. (21).

(10)* Calculate C.O.P. of the heat pump by Eq. (47).

(11)* Calculate the temperature of water returning from the heat pump, T_{Lh}, by Eq. (51).

(12) Calculate hourly auxiliary energy required (Q_{AUW} and Q_{AUX}) by Eqs. (41) and (42) for MODEL A or by Eqs. (42), (50), and (52) for MODEL B and C.

(13) Calculate cumulative auxiliary energy required, $\bar{Q}_{AUW}$ and $\bar{Q}_{AUX}$, by Eq. (43) and by Eq. (44).

(14) Determine F_1, F_2, and F_3 by Eq. (14), Eq. (15), and Eq. (16).

(15) Calculate the temperature of water in sections 1, 2, and 3 of the storage tank ($T_{s,1}$, $T_{s,2}$, and $T_{s,3}$) by Eqs. (11), (12), and (13)

(16) Calculate collector plate temperature, T_p, by Eq. (7).

(17) Repeat steps 2-17 for the next time interval.

Based on the equations derived in Chapter 3, the calculation results of the cumulative auxiliary energy requirements have units of Btu. Since all the auxiliary systems use electricity as their energy source, and since it was assumed that the conversion efficiency from electricity into heat was 1.0, the termal energy provided by the auxiliary systems (auxiliary energy required) was thus equivalent to the electrical energy consumed.

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*MODEL B and C only.

Since the simulation of MODEL B required the value of the heat pump horsepower to calculate the heat pump heating capacity, the size of the heat pump needed to be chosen before the simulation was made. Based on calculations of the cooling load for the hottest month (see Figure 5.1), it was found that a 3-ton (36,000 Btu per hour) air conditioner would be large enough to meet the cooling requirements of the house in the summer, therefore, a 3-ton air conditioner was used in MODEL A* and a 3-ton heat pump** was used in MODEL B* and MODEL C*.

In order to minimize the computer costs, only the first seven days of each month requiring heating (Oct. - Apr.) were simulated. The weather data of the first seven days of each month were punched in the computer cards and were read into the computer as the simulation basis. Continuous simulation was made by using the weather data at the end of one seven day period as the initial values for the beginning of the next seven day period. The performance of this seven day simulation was then multiplied by a factor (31/7 for October, December, January, and March; 30/7 for November and April; and 28/7 for February) to represent a whole month's result.

The computed monthly and yearly system performance for heating for MODEL A and MODEL B with collectors of 400, 600, 800, and 1000 ft^2 are shown in Tables 5.1 through 5.4 (MODEL C has the same performance as MODEL B.) Following these results, the yearly total heating load ($\bar{Q}_{AUW} + \bar{Q}_{AUX}$) fractions supplied by these collectors for both models are plotted against collector area in Figure 5.2. The figure shows clearly that MODEL B has a better thermal performance than MODEL A.

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*MODEL A uses an air conditioner for house cooling in the summer while MODEL B and MODEL C use the heat pump to cool the house.

**The size of the compressor for a 3-ton heat pump is 3 horsepower.

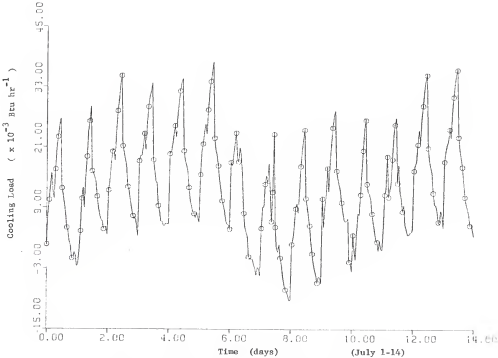

Figure 5.1 Simulated Cooling Load for the Hottest Month.

Table 5.1 The Monthly Cumulative Loads and Auxiliary Energy Required for MODEL A and MODEL B.

Month	Collector Area (ft^2)	Load (Btu x 10^{-6})			Auxiliary Energy Required (Btu x 10^{-6})					
		$\overline{Q}_{WL}$	$\overline{Q}_{HL}$	$\overline{Q}_{WL} + \overline{Q}_{HL}$	$\overline{Q}_{AUW}$		$\overline{Q}_{AUX}$		$\overline{Q}_{AUW} + \overline{Q}_{AUX}$	
					MODEL A	MODEL B	MODEL A	MODEL B	MODEL A	MODEL B
Oct.	400	1.41	10.14	11.55	0.31	0.69	6.40	1.58	6.71	2.27
	600				0.23	0.44	4.15	0.57	4.38	1.01
	800				0.20	0.31	2.52	0.26	2.72	0.57
	1000				0.16	0.21	1.35	0.10	1.51	0.31
Nov.	400	1.36	19.83	21.19	0.18	0.79	10.36	4.17	10.54	4.96
	600				0.12	0.40	5.25	0.95	5.37	1.35
	800				0.08	0.13	0.86	0.07	0.94	0.20
	1000				0.02	0.03	0.07	0.01	0.09	0.04
Dec.	400	1.41	31.79	33.20	0.38	1.25	24.94	17.99	25.32	19.24
	600				0.29	1.10	20.99	11.33	21.28	12.43
	800				0.24	0.83	17.31	6.55	17.55	7.38
	1000				0.20	0.55	12.63	3.95	12.83	4.50
Jan.	400	1.41	37.84	39.25	0.65	1.39	35.86	28.47	36.51	30.86
	600				0.54	1.36	34.48	26.00	35.02	27.36
	800				0.47	1.34	33.00	23.11	33.47	24.45
	1000				0.44	1.31	31.42	20.27	31.86	21.58
Feb.	400	1.27	29.12	30.39	0.43	1.19	24.30	18.13	24.73	19.32
	600				0.34	1.12	21.53	13.42	21.87	14.54
	800				0.28	1.03	18.69	9.84	18.97	10.87
	1000				0.25	0.93	15.94	6.96	16.19	7.89
Mar.	400	1.41	23.87	25.28	0.51	1.26	19.95	14.11	20.46	15.37
	600				0.40	1.18	17.92	11.00	18.32	12.18
	800				0.36	1.10	16.09	8.59	16.45	9.69
	1000				0.34	1.01	14.24	6.45	14.58	7.46
Apr.	400	1.36	16.55	17.91	0.47	1.11	13.74	8.02	14.21	9.13
	600				0.35	0.96	12.17	5.79	12.52	6.75
	800				0.28	0.82	10.77	3.96	11.05	4.78
	1000				0.26	0.70	9.42	2.63	9.68	3.33

Table 5.2 The Monthly System Performance for MODEL A and MODEL B.

Month	Collector Area (ft^2)	% Heating Load Supplied by Solar Energy					
		Hot Water		House Heating		Hot Water & House Heating	
		MODEL A	MODEL B	MODEL A	MODEL B	MODEL A	MODEL B
Oct.	400	78.0	51.1	36.9	84.4	41.9	80.3
	600	83.7	68.8	59.1	94.4	62.1	91.3
	800	85.8	78.0	75.1	97.4	76.5	95.1
	1000	88.7	85.1	86.7	99.0	86.9	97.3
Nov.	400	86.8	41.9	47.8	79.0	50.3	76.6
	600	91.2	70.6	73.5	95.2	74.7	93.6
	800	94.1	90.4	95.7	99.7	95.6	99.1
	1000	98.5	97.8	99.6	99.9	99.6	99.8
Dec.	400	73.0	11.3	21.6	43.4	23.7	42.1
	600	79.4	22.0	34.0	64.4	35.9	62.6
	800	83.0	41.1	45.6	79.4	47.1	77.8
	1000	85.8	61.0	60.3	87.6	61.4	86.5
Jan.	400	53.9	1.4	5.2	22.1	7.0	21.4
	600	61.7	3.6	8.9	31.3	10.8	30.3
	800	66.7	5.0	12.8	38.9	14.7	37.7
	1000	68.8	7.1	17.0	46.4	18.8	45.0
Feb.	400	66.1	6.3	16.6	37.7	18.6	36.4
	600	73.2	11.8	26.1	53.9	28.0	52.2
	800	78.0	18.9	35.8	66.2	37.6	64.2
	1000	80.3	26.8	45.3	76.1	46.7	74.0
Mar.	400	63.8	10.6	16.4	40.9	19.1	39.2
	600	71.6	16.3	24.9	53.9	27.5	51.8
	800	74.5	22.0	32.6	64.0	34.9	61.7
	1000	75.9	28.4	40.3	73.0	42.3	70.5
Apr.	400	65.4	18.4	17.0	51.5	20.7	49.0
	600	74.3	29.4	26.5	65.0	30.1	62.3
	800	79.4	39.7	34.9	76.1	38.3	73.3
	1000	80.9	48.5	43.1	84.1	46.0	81.4

Table 5.3 The Yearly Cumulative Heating Loads and Auxiliary Energy Required for MODEL A and MODEL B.

Collector Area (ft^2)	Load (Btu x 10^{-6}) $\overline{Q}_{WL}$	$\overline{Q}_{HL}$	$\overline{Q}_{WL} + \overline{Q}_{HL}$	Auxiliary Energy Required (Btu x 10^{-6}) $\overline{Q}_{AUW}$ MODEL A	$\overline{Q}_{AUW}$ MODEL B	$\overline{Q}_{AUX}$ MODEL A	$\overline{Q}_{AUX}$ MODEL B	$\overline{Q}_{AUW} + \overline{Q}_{AUX}$ MODEL A	$\overline{Q}_{AUW} + \overline{Q}_{AUX}$ MODEL B
400				2.93	7.68	135.55	93.47	138.48	101.15
600	9.63	169.14	178.77	2.27	6.56	116.49	69.06	118.76	75.62
800				1.91	5.56	99.24	52.38	101.15	57.94
1000				1.67	4.74	85.07	40.37	86.74	45.11

Table 5.4 The Yearly Heating System Performance for MODEL A and MODEL B.

Collector Area (ft^2)	% Heating Load Supplied by Solar Energy					
	Hot Water		House Heating		Hot Water & House Heating	
	MODEL A	MODEL B	MODEL A	MODEL B	MODEL A	MODEL B
400	69.6	20.3	19.9	44.7	22.5	43.4
600	76.4	31.9	31.1	59.2	33.6	57.7
800	80.2	42.3	41.3	69.0	43.4	67.6
1000	82.7	50.8	49.7	76.1	51.5	74.8

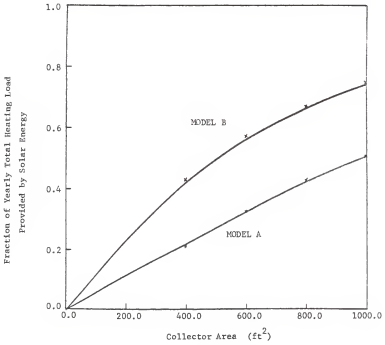

Figure 5.2 Fraction of Heating Load Supplied by Solar Energy as a Function of Collector Area.

Some dynamic results during the simulation are also plotted in Figures 5.3 through 5.28. In these figures, the units of time are days. Since only the first seven days of each month were simulated, the performance between 0 and 7 days stood for the performance for October, and the performance between 7 and 14 days in these figures stood for the performance for November, etc.

Figures 5.3 through 5.10 show the hourly auxiliary energy required for hot water heating for each model with collector areas of 400, 600, 800, and 1000 ft^2 during the simulation. Every figure shows that more auxiliary energy was needed in January (between days 21 and 28) than any other month and the least amount of auxiliary energy was needed in November (between days 7 and 14). By comparing these figures, we can see that as the collector area increases, the amount of auxiliary energy required decreases. The figures also show that the amount of auxiliary energy required for hot water heating for MODEL A is always less than the amount required for MODEL B for the same size collector. This is reasonable since the water temperature in the storage tank for MODEL A is most often higher than that for MODEL B during the simulation (as shown in Figures 5.27 and 5.28). The figures for MODEL B (Figures 5.7 through 5.10) also show that domestic hot water load is often supplied solely by the auxiliary heater because the auxiliary energy value shown in the figures often reaches its maximum (3784 Btu/hr). This indicates that the storage tank temperature is often below 70°F for MODEL B during the simulation.

Figures 5.11 through 5.18 show the hourly energy required for space heating for each model with each of the four collector areas. All the figures show that the highest hourly auxiliary energy required for space

heating happens at day 20 (in December). However, the month which requires the greatest amount of auxiliary energy is January since the area under the curve is the largest in this month. These figures also show that the amount of auxiliary energy required decreases as the collector area increases. By comparing the figures for the two models, MODEL B is shown to need less auxiliary energy than MODEL A for the same collector area. This indicates that the solar-assisted heat pump system has a better thermal performance than the conventional solar energy system.

The cumulative total auxiliary energy required was plotted against time in Figures 5.19 through 5.26. These figures show clearly that the larger the collector area is, the less is the amount of the auxiliary energy required. These figures also show that the amount of the cumulative auxiliary energy required for MODEL B is always less than that for MODEL A for the same collector size during the simulation.

The temperature of water in the storage tank during the simulation was also plotted against time in Figures 5.27 and 5.28. These figures show that the water temperature in the storage tank varies during the simulation between 100°F and 210°F for MODEL A and between 40°F and 210°F for MODEL B. The weather data during the simulation and the simulation programs for MODEL A and MODEL B are shown in the Appendix.

An entire month's simulation for MODEL A in January was also made. The attempt is to compare the performance of this month simulation with the previous estimated monthly performance which was calculated based on a seven day simulation. The results are shown in Table 5.5. It shows that the error of the estimated monthly performance is no more than 20%.

Table 5.5 Comparison Between the Simulated Monthly Performance and the Estimated Monthly Performance.
(MODEL A, Collector Area = 600 ft^2, January)

Item	Energy Quantities ($\times 10^{-6}$ Btu)			
	$\bar{Q}_{WL}$	$\bar{Q}_{HL}$	$\bar{Q}_{AUW}$	$\bar{Q}_{AUX}$
Simulated*	1.41	38.15	0.39	29.69
Estimated**	1.41	37.84	0.54	34.48
% Error***	0.0%	-0.8%	12.8%	17.9%

* An entire month's simulation results.

** The first seven day's simulation results multiplied by 31/7.

*** Based on simulated results = 100%.

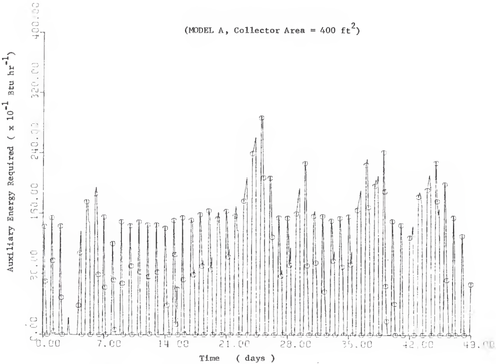

Figure 5.3 Simulation of Auxiliary Energy Required for Hot Water Heating.

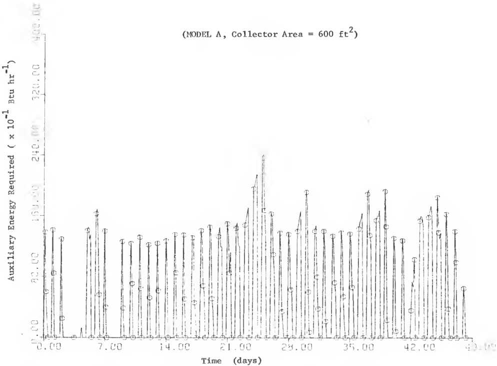

Figure 5.4 Simulation of Auxiliary Energy Required for Hot Water Heating.

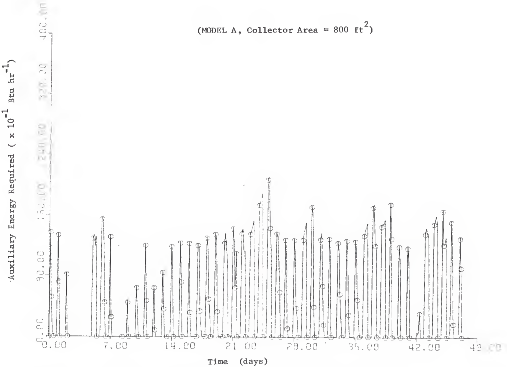

Figure 5.5 Simulation of Auxiliary Energy Required for Hot Water Heating.

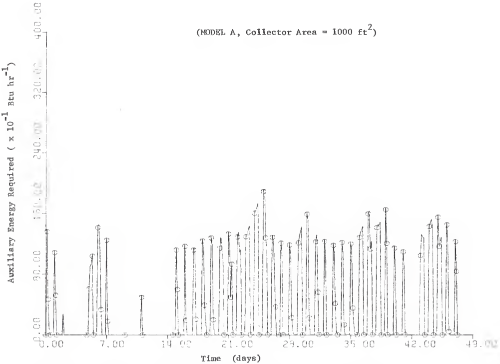

Figure 5.6 Simulation of Auxiliary Energy Required for Hot Water Heating.

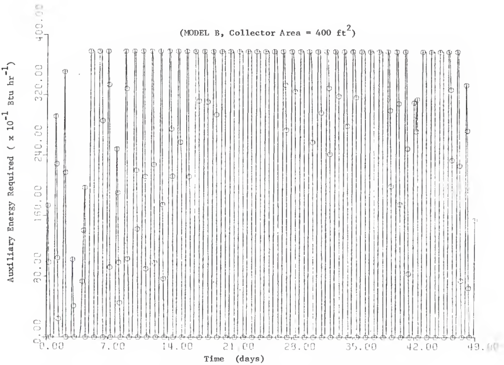

Figure 5.7 Simulation of Auxiliary Energy Required for Hot Water Heating.

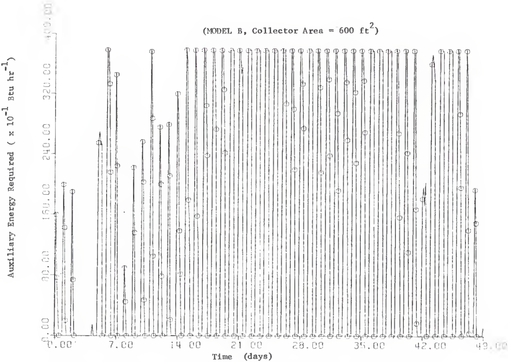

Figure 5.8 Simulation of Auxiliary Energy Required for Hot Water Heating.

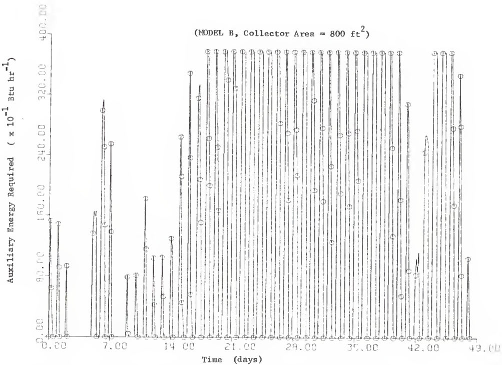

Figure 5.9 Simulation of Auxiliary Energy Required for Hot Water Heating.

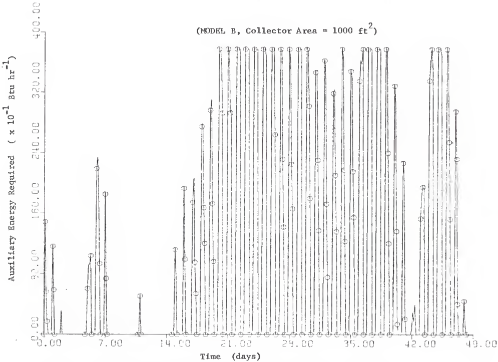

Figure 5.10 Simulation of Auxiliary Energy Required for Hot Water Heating.

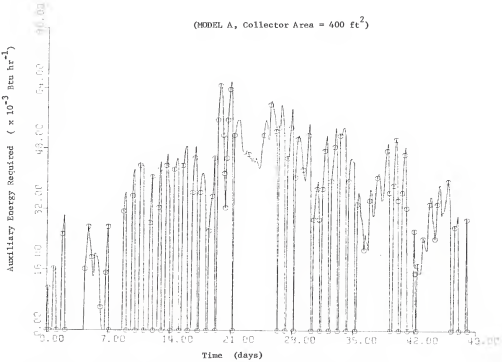

Figure 5.11 Simulation of Auxiliary Energy Required for House Heating.

(MODEL A, Collector Area = 600 ft^2)

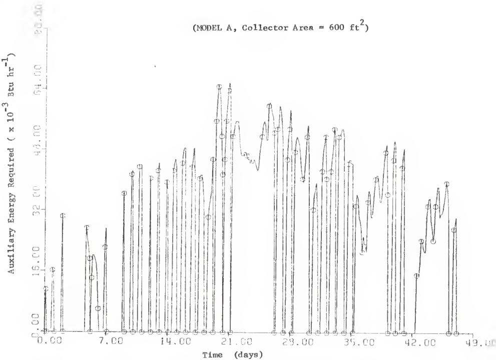

Figure 5.12 Simulation of Auxiliary Energy Required for House Heating.

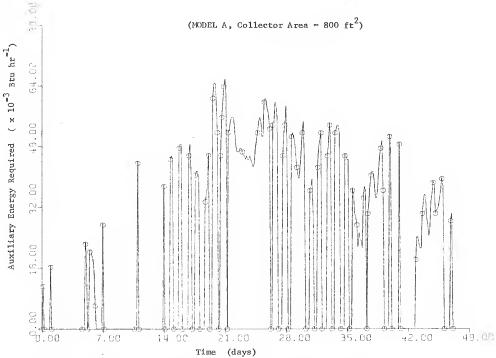

Figure 5.13 Simulation of Auxiliary Energy Required for House Heating.

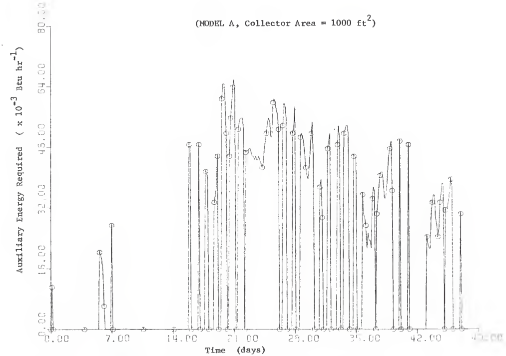

Figure 5.14 Simulation of Auxiliary Energy Required for House Heating.

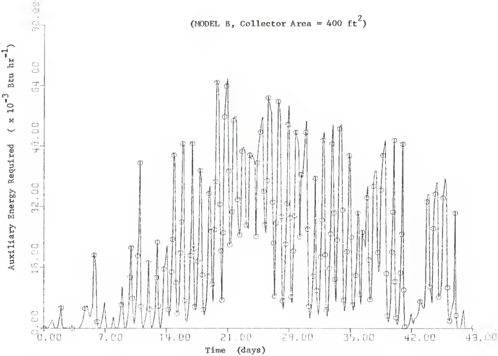

Figure 5.15 Simulation of Auxiliary Energy Required for House Heating.

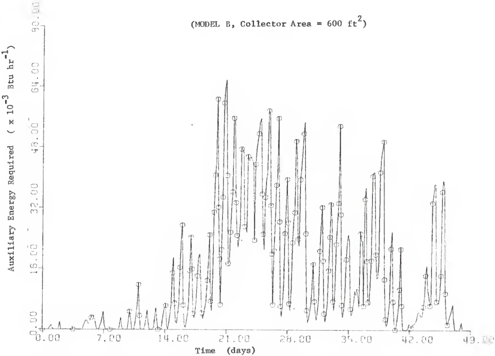

Figure 5.16 Simulation of Auxiliary Energy Required for House Heating.

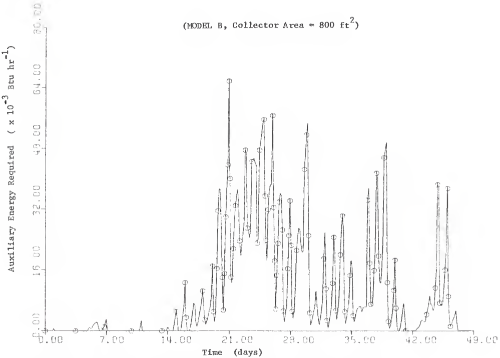

Figure 5.17 Simulation of Auxiliary Energy Required for House Heating.

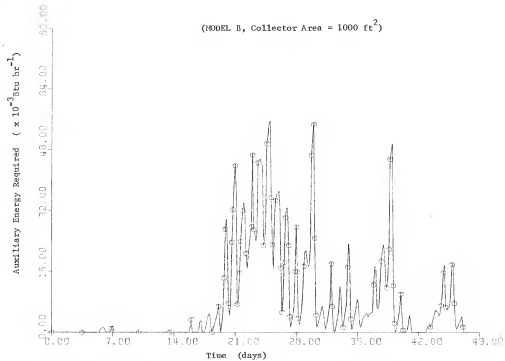

Figure 5.18 Simulation of Auxiliary Energy Required for House Heating.

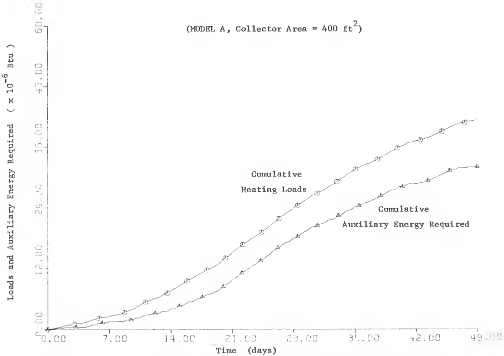

Figure 5.19 Cumulative Heating Loads and Cumulative Auxiliary Energy Required during the Simulation.

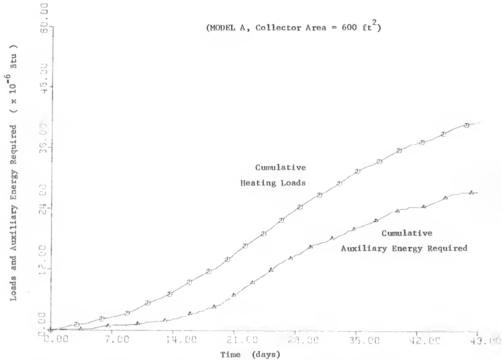

Figure 5.20 Cumulative Heating Loads and Cumulative Auxiliary Energy Required during the Simulation.

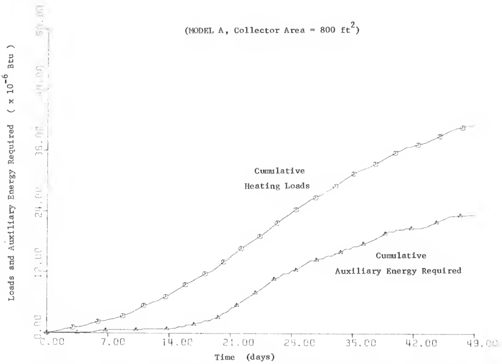

Figure 5.21 Cumulative Heating Loads and Cumulative Auxiliary Energy Required during the Simulation.

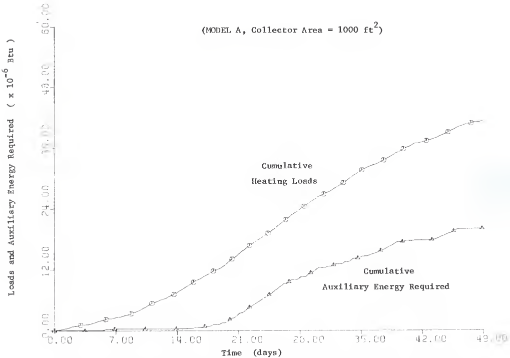

Figure 5.22 Cumulative Heating Loads and Cumulative Auxiliary Energy Required during the Simulation.

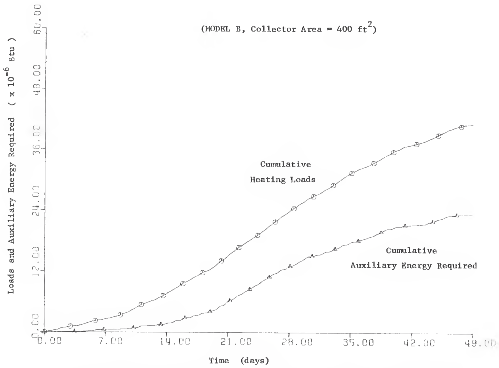

Figure 5.23 Cumulative Heating Loads and Cumulative Auxiliary Energy Required during the Simulation.

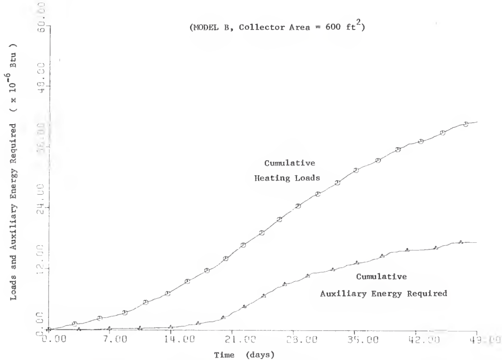

Figure 5.24 Cumulative Heating Loads and Cumulative Auxiliary Energy Required during the Simulation.

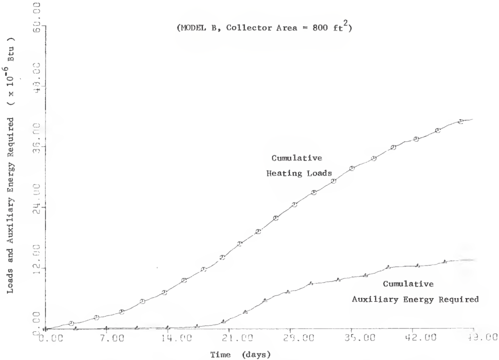

Figure 5.25 Cumulative Heating Loads and Cumulative Auxiliary Energy Required during the Simulation.

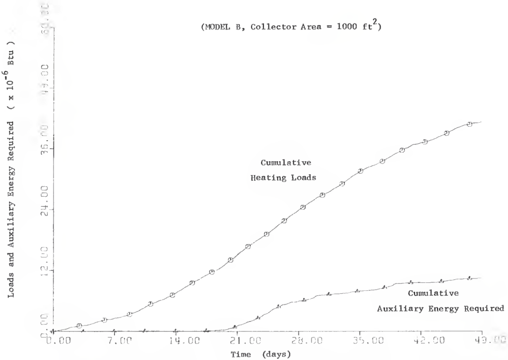

Figure 5.26 Cumulative Heating Loads and Cumulative Auxiliary Energy Required during the Simulation.

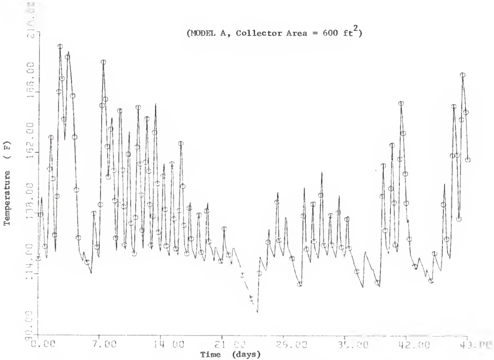

Figure 5.27 Temperature of Water in Section 1 of the Storage Tank during the Simulation.

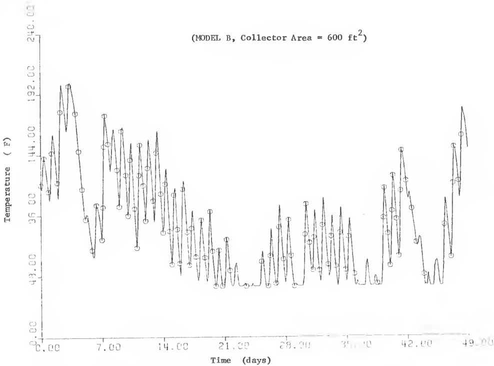

Figure 5.28 Temperature of Water in Section 1 of the Storage Tank during the Simulation.

COOLING RESULTS

The cooling result needed for the cost analysis is the cumulative auxiliary energy required (total electric energy consumed by the compressor) for each model in the summer. The amount of the cumulative auxiliary energy required was calculated by adding all the instantaneous hourly auxiliary energy required values which were estimated by the values of the C.O.P. and were calculated by Eq. (49) for MODEL A* and B** and by Eq. (59) for MODEL C***.

The simulation result showed that the cumulative auxiliary energy required for MODEL A and B was less than that for MODEL C (see Figure 5.29). This result was caused by the fact that the C.O.P. value for MODEL C during the simulation was mostly lower than that for MODEL A and MODEL B (see Figure 5.30). This means that the operation of a heat pump during the cooler summer night can not compensate for the lower suction temperature.

MODEL C has a lower suction temperature since it was assumed that the suction temperature for MODEL C was 10°F below the section 1 temperature of the storage tank. The storage tank temperature for MODEL C is varied during the simulation between 55°F and 40°F, so that, the averaged suction temperature for MODEL C is about 37°F instead of 55°F for MODEL A and B. This means that MODEL C will have better performance only when the temperature difference between daytime and nighttime is greater than 20°F. This is not the case for Manhattan's weather, so that MODEL C showed less efficiency. Table 5.6 clearly shows this fact.

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* An air conditioner was used for air-conditioning.

** The heat pump was designed to operate in the same manner as an air conditioner in the summer.

*** The heat pump was operated to cool the water in the storage tank during the night, the chilled water was used for cooling the house.

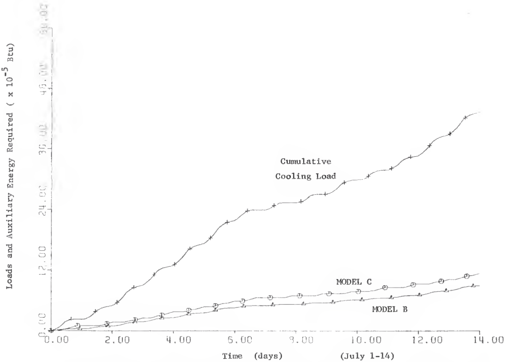

Figure 5.29 Cumulative Cooling Load and Cumulative Auxiliary Energy Required for MODEL B and for MODEL C during the Simulation.

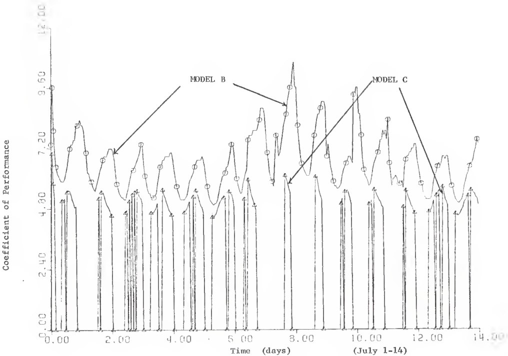

Figure 5.30 Values of C.O.P. for MODEL B and MODEL C during the Simulation.

Table 5.6 Values of the Heat Pump C.O.P. for MODEL B and MODEL C in the Averaged Summer Weather Condition in Manhattan. (1977)

Month	Averaged T_2^{*} (R)		Averaged T_1^{**} (R)		C.O.P.***	
	MODEL B	MODEL C	MODEL B	MODEL C	MODEL B	MODEL C
May	555	545	515	497	7.35	6.02
June	562	548	515	497	6.34	5.70
July	567	556	515	497	5.78	5.00
August	564	549	515	497	6.10	5.60
September	561	548	515	497	6.46	5.70

* For MODEL B - defined to be the average ambient temperature for the first 7 days of each month, each day from 9 AM to 9 PM , plus 20^oF.

For MODEL C - defined to be the average ambient temperature for the first 7 days of each month, each day from 0 AM to 7 AM, plus 20^oF.

**For MODEL B - 55^oF

For MODEL C - average operation temperature of the tank (47^oF) minus 10^oF.

*** Calculated by Eq.(43).

In Table 5.6, the averaged condensing temperatures were chosen as explained because the operation of the heat pump in MODEL B was mainly during the daytime from 9 AM to about 9 PM, and the operation of the heat pump in MODEL C was mainly from midnight to about 7 AM. It is evident from Table 5.5 that MODEL C would consume more auxiliary energy than MODEL B, therefore, MODEL C will not be discussed further.

In the next chapter, a cost analysis is made to compare the economic feasibility of MODEL A and MODEL B with a conventional heating and cooling system which uses an electric furnace for house heating, an electric hot water heater for hot water heating, and an air conditioner for house cooling. Since the result of interest is the dollar savings compared to the basis (operation and initial costs of a conventional system), and since it was assumed that a heat pump and an air conditioner consume the same amount of auxiliary energy in the summer, the exact amount of the auxiliary energy required for house cooling is not needed and was not calculated in this study.

CHAPTER 6. COST ANALYSIS

In this chapter a cost analysis is presented to show the economic feasibility of the two solar energy systems. The method of analysis was to determine the annual dollar savings of the systems. The term "annual dollar savings" refers to the amount of money that can be saved annually using a solar system for heating and cooling instead of using a conventional heating and cooling system. It was calculated by subtracting the annual costs of each system from that of a conventional system. The conventional system chosen in this study is one which uses an electric furnace for space heating, an electric heater for domestic hot water, and a 3-ton air conditioner for house cooling.

The major annual costs of MODEL A and MODEL B include: the annual cost of owning the collector, storage unit and associated controls, pumps, piping and the like; the annual cost of owning the auxiliary equipment, an electric furnace, an electric hot water heater, and an air conditioner or a heat pump; the yearly cost of operating the system; and the yearly cost of maintenance. The annual cost of ownership includes costs associated with the initial investment, that is, interest on the investment and its repayment over a specified number of years related to its life time. The sum of these is usually taken as a fixed percentage of investment each year; for example, for a 20 year amortization and 8% interest rate, the annual cost is 0.10185 of the investment.

Operating costs are primarily due to power requirements for the electric furnace, the electric hot water heater, the air conditioner or the heat pump, and for pumping water and moving air in the system, summed over the yearly

operating time of the system. Maintenance costs include repairs, replacement of glass in collectors, or any other costs of keeping the system in operating condition.

The annual costs of MODEL A and MODEL B, in dollars per year, can be formulated as follows:

For MODEL A

$$C_{MA} = (C_c A_c + C_{st} + C_e)(I_n) + (C_f + C_w + C_{ai})(I'_n) + (P_{A,h} + P_{A,c})(C_r) + C_m \quad (60)$$

For MODEL B

$$C_{MB} = (C_c A_c + C_{st} + C_e)(I_n) + (C_f + C_w + C_{hp})(I'_n) + (P_{B,h} + P_{B,c})(C_r) + C_m \quad (61)$$

For the conventional system, the annual costs are the sum of the following: the annual cost of owning an electric furnace, an electric hot water heater, an air conditioner, and the annual power requirements of operating them. It is formulated as:

$$C_{CON} = (C_f + C_w + C_{ai})(I'_n) + (P_{CON,h} + P_{CON,c})(C_r) \quad (62)$$

Since it has been assumed that $P_{A,c} = P_{B,c} = P_{CON,c}$, the annual dollar savings for MODEL A and MODEL B can be calculated by:

For MODEL A

$$(DS)_A = C_{CON} - C_{MA} = (P_{CON,h} - P_{A,h})(C_r) - (C_c A_c + C_{st} + C_e)(I_n) - C_m \quad (63)$$

For MODEL B

$$(DS)_B = C_{CON} - C_{MB} = (P_{CON,h} - P_{B,h})(C_r) - (C_c A_c + C_{st} + C_e)(I_n) - (C_{hp} - C_{ai})(I'_n) - C_m \quad (64)$$

The annual dollar savings for MODEL A and MODEL B for collector areas of 400, 600, 800, and 1000 ft^2 at electrical rates of \$0.02, 0.03, 0.04, and 0.05 per KWh were calculated and are presented in Table 6.1. The calculations were based on Eq. (63) and Eq. (64) together with the following assumed values:

Table 6.1 Annual Dollar Savings for MODEL A and MODEL B for Collector Areas of 400, 600, 800, and 1000 ft^2 at Power Rates of \$0.02, 0.03, 0.04, and 0.05/KWh.

Power Rate (\$/KWh)	Collector Area (ft^2)	Annual Dollar Savings (\$)	
		MODEL A	MODEL B
0.02	400	-247	-130
	600	-317	-166
	800	-400	-248
	1000	-501	-359
0.03	400	-109	118
	600	-121	156
	800	-151	126
	1000	-211	53
0.04	400	30	365
	600	75	479
	800	96	500
	1000	78	465
0.05	400	168	613
	600	271	801
	800	343	874
	1000	368	876

The capital cost per unit area of collector*, $C_c = \$8.1/ft^2$

The capital cost of storage*, $C_{st} = (\$0.082/lb)(lb\ water\ stored)$

The capital cost of additional equipment*, $C_e = \$500$

Annual factor of investment, I_n and $I'_n = 0.10185$***

The cost difference between a heat pump and an air conditioner, $C_{hp} - C_{ai} = \$1000$**

The annual cost of maintenance, $C_m = \$100$

The value of $P_{A,h}$ and $P_{B,h}$ in the above calculations come from the appropriate terms of the total auxiliary energy required in Table 5.3 by converting their units from Btu to KWh. This is reasonable since all the auxiliary energy comes from electricity. The value of $P_{CON,h}$ is the sum of the total heating loads plus 6.81 million Btu of hot water load in the summer.**** The term was added since the solar systems were assumed to meet all the domestic hot water load in the summer.

The annual dollar savings of MODEL A and MODEL B was plotted against collector area in Figure 6.1. It can be seen in Figure 6.1 that MODEL B is better than MODEL A for the costs assumed in this study and that the electrical rate must be at about \$0.036 and at \$0.025 per KWh respectively before any dollar savings for MODEL A and for MODEL B can occur. The optimum collector size for MODEL A at \$0.036/KWh is about 600 ft^2, and that for MODEL B at \$0.025/KWh is about 500 ft^2. Since the current winter electrical rate for Manhattan is \$0.030/KWh, we can see that the solar-assisted heat pump system

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* From reference (11).

** From reference (25).

*** Corresponding to 20 year amoratization with 8% interest rate.

**** $(3784\ Btu/hr)(12\ hr/day)(30\ day/month)(5\ month) = (6.81 \times 10^6\ Btu)$

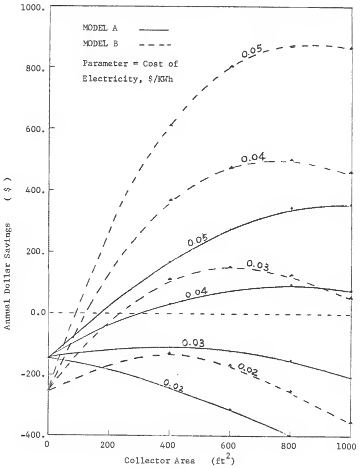

Figure 6.1 Annual Dollar Savings for MODEL A and MODEL B.

(MODEL B) is now economically competitive with a conventional all electric system.

Figure 6.1 also shows that the dollar savings for solar energy systems is not a strong function of collector area in a large range about the optimum. Thus, small errors in the estimate of the long-term thermal performance of solar energy systems will have little effect on the resultant optimum collector size.

In order to see how much the assumed annual factor of investment will effect the results, a calculation was made using different values of the annual factor of investment (I_n and I'_n) for a power cost of \$0.04/KWh. The calculation was again based on Eqs. (63) and (64), the results are shown in Figure 6.2. In the figure, the parameter value of 0.149 corresponds to 10 year amoratization with 8% interest rate while the value of 0.250 corresponds to 5 year amoratization with 8% interest rate. From the figure, we can see that the assumed annual factor of investment has a big effect upon both the amount of dollar savings and the optimum collector size. The figure shows that as the annual factor of investment increases, the amount of dollar savings decreases and the optimum collector size also decreases. MODEL B is still better than A.

Although no calculation was made, different assumed costs for the collector and the storage tank would also have a great effect on the dollar savings and the optimum collector area. This is also true for other assumed values. Since these values are hard to justify because the industry is not fully developed, the results may not give accurate values but they do indicate an economic trend for these solar energy systems.

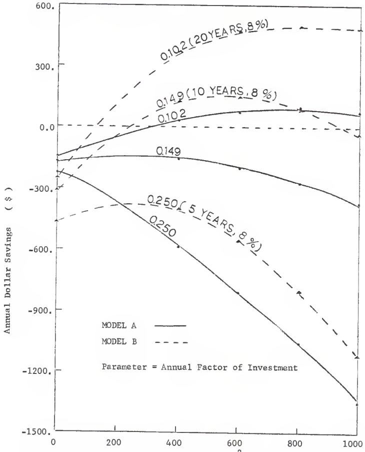

Figure 6.2 Annual Dollar Savings for MODEL A and MODEL B for Power Cost at \$0.04/KWh.

CHAPTER 7. CONCLUSION

The economic feasibility of direct use of solar energy for residential heating and cooling in Manhattan, Kansas has been studied in this work. Three solar heating and cooling systems were modeled and simulated using the 1976-1977 hourly weather bureau data of Manhattan. The simulation programs were written using the Continuous System Modeling Program and the simulation results were reported in the previous chapters.

The simulation results of heating (see Tables 5.3 and 5.4) show that the solar-assisted heat pump system (MODEL B) has a better thermal performance than the conventional solar energy heating system (MODEL A). The simulation results of cooling (see Figure 5.29) show that the heat pump operation in MODEL C[*] is less efficient than in MODEL B[**]. The Continuous System Modeling Program (CSMP) which was used in this study has proved itself to be a powerful computer language in simulating the solar energy systems. Although it uses more computer time for compiling, programming using CSMP is much easier than using other computer languages.

Under the assumed costs for solar equipment and other fees, the cost analysis made in the previous chapter (see Table 6.1 and Figure 6.1) has resulted the following conclusions:

(1) The solar-assisted heat pump system is superior to the conventional solar energy system. The result of the cost analysis shows that the amount of dollar savings for MODEL B is greater than that for MODEL A at any electrical rate.

---

[*] The heat pump operation was designed to cool the water in the storage tank during the summer night, the chilled water was used to cool the house.

[**] The heat pump cooling operation was assumed the same as an air conditioner.

(2) Compared with the conventional heating and cooling system which uses electricity as its energy source, MODEL B with an assumed equipment life of 20 years will save money once the electrical rate is above \$0.025/KWh. At this electrical rate, the optimum collector size for MODEL B would be about 500 ft^2, and for this collector size, the total heating load supplied by solar energy would be about 51%.

(3) For MODEL A with an equipment life of 20 years to compare favorably with the conventional system, the electrical rate must be about \$0.036/KWh before a dollar savings can occur. At this electrical rate, the optimum collector size would be about 600 ft^2, and for this collector size, the total heating load supplied by solar energy would be about 34%.

(4) The greater the electrical rate is, the larger will be the optimum collector size. This can be seen from Figure 6.1. For example, at electrical rate of \$0.03/KWh, the figure shows the optimum collector size for MODEL B is 600 ft^2, but at \$0.04/KWh, it shows the optimum collector size becomes 800 ft^2.

(5) With an assumed equipment life of 20 years, MODEL A shows that it has negative amount of dollar savings while MODEL B shows that it has positive amount of dollar savings at the present winter electrical rate of \$0.030/KWh for Manhattan. The results show that the optimum collector size at this current electrical rate is 400 ft^2 for MODEL A and 600 ft^2 for MODEL B, and the annual dollar savings is about \$-109 for MODEL A and \$156 for MODEL B (see Figure 6.1).

(6) The assumed equipment life has a big effect on the values of dollar savings for both models. As the assumed life decreases, the value of the annual factor of investment (I_n and I'_n) will increase, thus decreasing the amount of dollar savings for both models. The calculated results still show that MODEL B is better than MODEL A for every assumed equipment life value (see Figure 6.2).

Under the assumed costs for solar equipment and other fees, the solar-assisted heat pump system shows that it is now economically feasible compared to the conventional system which uses electricity as its energy source. However, this does not mean that the solar energy system is more attractive than a conventional heating and cooling system at present, since most of the residences in the United States use much cheaper energy sources, - i.e., gas and oil, for heating. The prospects for use of a solar energy heating and cooling system for a residence are thus heightened by the impending shortages and cost increases of oil and gas.

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APPENDIX A

Simulation Program of Heating
for MODEL A

```
**************************************************************
*                                                            *
*  MODEL A - SOLAR ENERGY HEATING AND CONVENTIONAL COOLING   *
*                                                            *
*                    ( HEATING PORTION )                     *
*                                                            *
**************************************************************
INITIAL
CONST ZE=0.0,S=49.0,EP=0.95,EG=0.88,UBE=0.16,W=0.323,...
D=0.069,DI=0.049,HFI=265.0,K=121.4,DEL=0.00164,T=0.9,...
ALFA=0.9,N=2,A2= 7000.0,HL1=11.615,HL2=8.49,HL3=8.49,...
R1=1.0,AREA=600.0,A3=1000.0
/      DIMENSION X(1176),Y(1176),Z(1176),U(7)
FIXED I,J,IJJ
INCON IC=120.0
NOSORT
   11 FORMAT (7F8.6)
   10 FORMAT (12F5.3)
      READ (5,10) (X(I),I=1,1176)
      READ (5,10) (Y(I),I=1,1176)
      READ (5,10) (Z(I),I=1,1176)
      READ (5,11) (U(I),I=1,7)
DYNAMIC
DYNAMIC
NOSORT
      IF (TIME.EQ.ZE) GO TO 25
      GO TO 20
   25 WII=10.0
      TA=27.0
      MAXIN=0.0
      TP=IC
      TOTAL=0.0
      AUX=0.0
      AUXIL=0.0
      AUXW=0.0
      AAUX=0.0
      TOTAL1=0.0
      TOTAL2=0.0
      CH=0.0
      TI1=IC
      TI2=IC
      TI3=IC
      CP=AREA*12.5
      A1=AREA*10.0
      GO TO 30
   20 TE=TIME+0.99
      J=TE
      WII=Z(J)
      TA=Y(J)
      TTE=J
      LS=TTE/168.0+0.9989
      IJJ=LS
      CDD=U(IJJ)
      CD=COS(CDD)
      SD=SIN(CDD)
      AH=TTE*0.2618
      CH=SIN(AH)
      IF (CH.LE.0.) GO TO 2?
```

```
      SB=0.777*CD*CF+0.629*SD
      CB=(1.0-SB**2)**0.5
      SF=CD*(1.0-CH**2)**0.5/CB
      CF=(1.0-SF**2)**0.5
      CC=0.7547*CB*CF+0.656*SP
      MAN=X(J)*3.68663*CC/SB
      MAXIN=INSW(MAN,ZE,MAN)
      GO TO 30
   28 MAXIN=0.0
      SB=0.0
      CB=0.0
      CF=0.0
      SF=0.0
      CC=0.0
   30 CONTINUE
SORT
HHWW=1.004+0.3*W11
FFSS=(1.0-0.227*HHWW+0.016*HHWW**2)*(1.0+0.091*N)
C=365.9*(1.0-0.00883*S+0.0001298*S**2)
TKK=TA+460.0
TUU=TP+460.0
TUUTKK=ABS(TUU-TKK)
UL=1.0/((3.829*N)/((C/TUU)*((TUUTKK/(N+FFSS))**0.33))+...
 1.0/HHWW)+0.001712*(TUU**2+TKK**2)*0.001*(TUU+TKK)*0.001/...
 (1.0/(EP+0.05*N*(1.0-EP))+(2.0*N+FFSS-1.0)/EC-N)+UBL
D4=UL/(K*DEL)
M=D4**0.5
QQ=M*(W-D)/2.0
F=(1.0/QQ)*TANH(QQ)
FP=1.0/(UL*W*(1.0/(UL*(D+(W-D)*F))+1.0/(3.1416*D1*HF1)))
FR=A1*(1.0-EXP(-FP*UL*AREA/A1))/(AREA*UL)
TOUT=T13+AREA*FR*(MAXIN*T*ALFA-UL*(T13-TA))/A1
XX=TOUT-T13
YY=INSW(XX,ZE,R1)
LOA=1014.0*(70.0-TA)
LOAD=INSW(LOA,ZE,LOA)
IND=INSW(CH,ZE,R1)
LOAD1=3784.0*IND
NOSORT
      IF (TOLT.GE.210.0) GO TO 310
      GO TO 320
  310 YY=0.0
  320 CONTINUE
      WAT=1.0
      IF(T11.LE.70.0) GO TO 500
      GO TO 510
  500 WAT=0.0
  510 CONTINUE
      THOT=T11-10.0
      TWA=T11-47.3*(THOT-60.0)/A3
      TRB=T11-LOAD/A2
      IF (TIME.EQ.ZE) GO TO 83
      IF (KEEP.EQ.0) GO TO 83
      IF (KEEP.EQ.1) GO TO 82
      GO TO 83
   82 TOTAL=TOTAL+LOAD*DELT
      TOTAL1=TOTAL1+LOAD1*DELT
   83 IF (TRB.LE.120.) GO TO 51
      AUXIL=0.0
```

```
   80 CONTINUE
SORT
DTI1=(A1*YY*(TOUT-TI1)+A2*(TI2-TI1)-HL1*(TI1-40.0)+A3*...
 (TI2-TI1)*WAT*IND)/CP*3.0
TI1=INTGRL(IC,DTI1)
DTI2=(A1*YY*(TI1-TI2)+A2*(TI3-TI2)-HL2*(TI2-40.0)+A3*...
 (TI3-TI2)*WAT*IND)/CP*3.0
TI2=INTGRL(IC,DTI2)
DTI3=(A1*(TI2-TI3)*YY+A2*(TRP-TI3)-HL3*(TI3-40.0)+A3*...
 (TWA-TI3)*WAT*IND)/CP*3.0
TI3=INTGRL(IC,DTI3)
NOSORT
      GO TO 100
   51 IF (TIME.EQ.ZE) GO TO 58
      IF (KEEP.EQ.0) GO TO 58
      IF (KEEP.EQ.1) GO TO 55
      GO TO 58
   55 AUX=AUX+LOAD*DELT
      AUXIL=LOAD
   58 CONTINUE
SORT
DTI1=(A1*YY*(TOUT-TI1)-HL1*(TI1-40.0)+A3*(TI2-TI1)*...
 WAT*IND)/CP*3.0
TI1=INTGRL(IC,DTI1)
DTI2=(A1*YY*(TI1-TI2)-HL2*(TI2-40.0)+A3*(TI3-TI2)*...
 WAT*IND)/CP*3.0
TI2=INTGRL(IC,DTI2)
DTI3=(A1*YY*(TI2-TI3)-HL3*(TI3-40.0)+A3*(TWA-TI3)*...
 WAT*IND)/CP*3.0
TI3=INTGRL(IC,DTI3)
NOSORT
  100 TP=TI3+(MAXIN*ALFA*T-UL*(TI3-TA))/UL*(1.0-FR/FP)
      IF (TIME.EQ.ZE) GO TO 90
      IF(KEEP.EQ.0) GO TO 92
      IF (KEEP.EQ.1) GO TO 95
      GO TO 92
   95 FAUXWA=47.3*(140.0-THOT)*WAT*IND+3784.0*(1.0-WAT)*IND
      AUXWA=INSW(FAUXWA,ZE,FAUXWA)
      AUXW=AUXW+AUXWA*DELT
      AAUX=AUX+AUXW
      TOTAL2=TOTAL+TOTAL1
      PERC=(1.0- AUX/TOTAL)*100
      PERC1=(1.0-AUXW/TOTAL1)*100.0
      PERC2=(1.0-AAUX/TOTAL2)*100.0
      SOLAR=SOLAR+AREA*MAXIN*DELT
      IF (SOLAR.EQ.ZE) GO TO 91
      USE=(TOTAL-AUX)/SOLAR*100.0
      GO TO 92
   91 USE=0.0
      GO TO 92
   90 PERC=0.0
      PERC1=0.0
      PERC2=0.0
      SOLAR=0.0
      USE=0.0
   92 CONTINUE
      TTIME=TIME/24.0
SORT
PAGE XYPLOT,HEIGHT=5,WIDTH=7
```

```
OUTPUT TTIME(0.0,49.0),TA
OUTPUT TTIME(0.0,49.0),WII
OUTPUT TTIME(0.0,49.0),MAXIN
OUTPUT TTIME(0.0,49.0),TII,TI2,TI3
PAGE GROUP
OUTPUT TTIME(0.0,49.0),TI1
OUTPUT TTIME(0.0,49.0),PERC(0.0,100.0)
PAGE GROUP=(0.0,100.0)
OUTPUT TTIME(0.0,49.0),PERC1(0.0,100.0)
PAGE GROUP=(0.0,100.0)
OUTPUT TTIME(0.0,49.0),PERC2(0.0,100.0)
PAGE GROUP=(0.0,100.0)
OUTPUT TTIME(0.0,49.0),TOTAL,AUX
PAGE GROUP
OUTPUT TTIME(0.0,49.0),TOTAL1,AUXW
PAGE GROUP
OUTPUT TTIME(0.0,49.0),TOTAL2,AAUX
PAGE GROUP
OUTPUT TTIME(0.0,49.0),LOAD1,AUXWA
PAGE GROUP
OUTPUT TTIME(0.0,49.0),AUXWA(0.0,4000.0)
PAGE GROUP=(0.0,4000.0)
OUTPUT TTIME(0.0,49.0),LOAD
PAGE GROUP=(0.0,80000.0)
OUTPUT TTIME(0.0,49.0),AUXIL(0.0,80000.0)
PAGE GROUP=(0.0,80000.0)
OUTPUT TTIME(0.0,49.0),LOAD,AUXIL
PAGE GROUP=(0.0,80000.0)
OUTPUT TTIME(0.0,49.0),FP
OUTPUT TTIME(0.0,49.0),FR
OUTPUT TTIME(0.0,49.0),UL
TITLE SYSTEM DESCRIPTION
PRINT USE,PERC,MAXIN,LOAD,TOTAL,TII,TKB,AUX,YY,XX,AUXIL,...
CD,TI2,TI3,TOUT,SB,CB,SF,CF,CC,TIL,DTI1,DTI2,DTI3,FR,...
AUXWA,AUXW,AAUX,TOTAL1,TOTAL2,PERC1,PERC2,IND,LOAD1,...
SOLAR,TKK,TUU,TUUTKK,FFSS,HHWW,UL,U4,M,CC,F,FP,TWA,WAT
TIMER OUTDEL=1.0,FINTIM=1176.0,DELT=0.25,PRDEL=1.0
METHOD RKSFX
END
```

APPENDIX B

Simulation Program of Heating for MODEL B

```
***********************************************************************
*                                                                     *
* MODEL B - SOLAR-ASSISTED HEAT PUMP FOR HEATING AND COOLING         *
*                                                                     *
*                        ( HEATING PORTION )                          *
*                                                                     *
***********************************************************************
INITIAL
CONST ZE=0.0,S=49.0,EP=0.55,EG=0.68,ULE=0.16,W=0.328,...
D=0.069,DI=0.049,HFI=265.0,K=121.4,DEL=0.00164,T=0.9,...
ALFA=0.9,N=2,A2= 7000.0,HL1=11.615,HL2=8.49,HL3=8.49,...
RI=1.0,AREA=600.0,A3=1000.0
/     DIMENSION X(1176),Y(1176),Z(1176),U(7)
FIXED I,J,IJJ
INCON IC=120.0
NOSORT
   11 FORMAT (7F8.6)
   10 FORMAT (12F5.3)
      READ (5,10) (X(I),I=1,1176)
      READ (5,10) (Y(I),I=1,1176)
      READ (5,10) (Z(I),I=1,1176)
      READ (5,11) (U(I),I=1,7)
DYNAMIC
NOSORT
      IF (TIME.EQ.ZE) GO TO 25
      GO TO 20
   25 WII=10.0
      TA=27.0
      MAXIN=0.0
      TP=IC
      TOTAL=0.0
      TOTAL1=0.0
      TOTAL2=0.0
      AUX=0.0
      AUXW=0.0
      AAUX=0.0
      CH=0.0
      AUXI=0.0
      AUXIL3=0.0
      TI3=IC
      TI2=IC
      TI1=IC
      CP=AREA*12.5
      A1=AREA*10.0
      GO TO 30
   20 TE=TIME+0.99
      J=TE
      TA=Y(J)
      WII=Z(J)
      TTE=J
      LS=TTE/168.0+0.9999
      IJJ=LS
      CDD=U(IJJ)
      CD=COS(CDD)
      SD=SIN(CDD)
      AH=TTE*0.2618
      CH=SIN(AH)
      IF (CH.LE.0.) GO TO 29
```

```
      SB=0.777*CD*CH+0.629*SD
      CB=(1.0-SB**2)**0.5
      SF=CD*(1.0-CH**2)**0.5/CB
      CF=(1.0-SF**2)**0.5
      CC=0.7547*CB*CF+0.656*SB
      MAN=X(J)*3.68663*CC/SB
      MAXIN=INSW(MAN,ZE,MAN)
      GO TO 30
   28 MAXIN=0.0
      SB=0.0
      CB=0.0
      CF=0.0
      SF=0.0
      CC=0.0
   30 CONTINUE
SORT
HHWW=1.004+0.3*WI1
FFSS=(1.0-0.227*HHWW+0.016*HHWW**2)*(1.0+0.091*N)
C=365.9*(1.0-0.00883*S+0.0001298*S**2)
TKK=TA+460.0
TUU=TP+460.0
TUUTKK=ABS(TUU-TKK)
UL=1.0/((3.829*N)/((C/TUU)*((TUUTKK/(N+FFSS))**0.33))+...
 1.0/HHWW)+0.001712*(TUU**2+TKK**2)*0.001*(TUU+TKK)*0.001/...
 (1.0/(EP+0.05*N*(1.0-EP))+(2.0*N+FFSS-1.0)/EC-N)+UBE
D4=UL/(K*DEL)
M=D4**0.5
QQ=M*(W-D)/2.0
F=(1.0/QQ)*TANH(QQ)
FP=1.0/(UL*W*(1.0/(UL*(D+(W-D)*F))+1.0/(3.1416*DI*HFI)))
FR=A1*(1.0-EXP(-FP*UL*AREA/A1))/(AREA*UL)
TOUT=T13+AREA*FR*(MAXIN*T*ALFA-UL*(T13-TA))/A1
XX=TOUT-T13
YY=INSW(XX,ZE,R1)
LOA=1014.0*(70.0-TA)
LOAD=INSW(LOA,ZE,LOA)
IND=INSW(CH,ZE,R1)
LOAD1=3784.0*IND
NOSORT
      IF (TOUT.GE.210.0) GO TO 310
      GO TO 320
  310 YY=0.0
  320 CONTINUE
      WAT=1.0
      IF(T11.LE.70.0) GO TO 500
      GO TO 510
  500 WAT=0.0
  510 CONTINUE
      THOT=T11-10.0
      TRA=T11-47.3*(THOT-60.0)/A3
      TRB=T11-LOAD/A2
      IF (TIME.EQ.ZE) GO TO 83
      IF (KEEP.EQ.0) GO TO 83
   82 TOTAL=TOTAL+LOAD*DELT
      TOTAL1=TOTAL1+LOAD1*DELT
   83 IF (TRB.LE.120.) GO TO 51
      ICOP=0.0
      COP=0.0
      AUXIL3=0.0
```

```
      AUXI=0.0
   80 CONTINUE
SORT
DTI1=(A1*YY*(TOUT-TI1)+A2*(TI2-TI1)-HL1*(TI1-40.0)+A3*...
 (TI2-TI1)*WAT*IND)/CP*3.0
TI1=INTGRL(IC,DTI1)
DTI2=(A1*YY*(TI1-TI2)+A2*(TI3-TI2)-HL2*(TI2-40.0)+A3*...
 (TI3-TI2)*WAT*IND)/CP*3.0
TI2=INTGRL(IC,DTI2)
DTI3=(A1*(TI2-TI3)*YY+A2*(TRB-TI3)-HL3*(TI3-40.0)+A3*...
 (TWA-TI3)*WAT*IND)/CP*3.0
TI3=INTGRL(IC,DTI3)
NOSORT
      GO TO 100
   51 ICOP=580.0/(120.0-TI1+10.0)
      COP=0.53*ICOP
      CAPA=7633.65*COP
      IF (LOAD.LE.CAPA) GO TO 200
      TRB=TI1-7633.65*(COP-1.0)/A2
      IF (TI1.LE.40.0) GO TO 60
      IF (TIME.EQ.ZE) GO TO 80
      IF (KEEP.EQ.0) GO TO 80
   55 AUX=AUX+CAPA/COP*DELT+(LOAD-CAPA)*DELT
      AUXI=CAPA/COP+(LOAD-CAPA)
      GO TO 210
  200 TRB=TI1-LOAD*(1.0-1.0/COP)/A2
      IF (TI1.LE.40.0) GO TO 60
      IF (TIME.EQ.ZE) GO TO 80
      IF (KEEP.EQ.0) GO TO 80
   56 AUX=AUX+LOAD/COP*DELT
      AUXI=LOAD/COP
  210 AUXIL3=0.0
      GO TO 80
   60 IF (TIME.EQ.ZE) GO TO 68
      TRB=0.0
      IF (KEEP.EQ.0) GO TO 68
   65 AUX=AUX+LOAD*DELT
      AUXI=LOAD
      AUXIL3=LOAD
   68 CONTINUE
SORT
DTI1=(A1*YY*(TOUT-TI1)-HL1*(TI1-40.0)+A3*(TI2-TI1)*...
 WAT*IND)/CP*3.0
TI1=INTGRL(IC,DTI1)
DTI2=(A1*YY*(TI1-TI2)-HL2*(TI2-40.0)+A3*(TI3-TI2)*...
 WAT*IND)/CP*3.0
TI2=INTGRL(IC,DTI2)
DTI3=(A1*YY*(TI2-TI3)-HL3*(TI3-40.0)+A3*(TWA-TI3)*...
 WAT*IND)/CP*3.0
TI3=INTGRL(IC,DTI3)
NOSORT
  100 TP=TI3+(MAXIN*ALFA*T-UL*(TI3-TA))/UL*(1.0-FR/FP)
      IF (TIME.EQ.ZE) GO TO 90
      IF(KEEP.EQ.0) GO TO 92
      IF (KEEP.EQ.1) GO TO 95
      GO TO 92
   95 FAUXWA=47.3*(140.0-THOT)*WAT*IND+3784.0*(1.0-WAT)*IND
      AUXWA=INSW(FAUXWA,ZI,FAUXWA)
      AUXW=AUXW+AUXWA*DELT
```

```
      AAUX=ALX+ALXW
      TOTAL2=TOTAL+TOTAL1
      PERC=(1.0- AUX/TOTAL)*100
      PERC1=(1.0-AUXW/TOTAL1)*100.0
      PERC2=(1.0-AAUX/TOTAL2)*100.0
      SOLAR=SOLAR+AREA*MAXIN*DELT
      IF(SOLAR.EQ.ZE) GO TO 91
      USF=TOTAL*PERC/SOLAR
      GO TO 92
   91 USF=0.0
      GO TO 92
   90 PERC=0.0
      PERC1=0.0
      PERC2=0.0
      SOLAR=0.0
      USF=0.0
   92 CONTINUE
      TTIME=TIME/24.0
SORT
PAGE XYPLOT,HEIGHT=5,WIDTH=7
OUTPUT TTIME(0.0,49.0),TI1,TI2,TI3
PAGE GROUP
OUTPUT TTIME(0.0,49.0),TI1
OUTPUT TTIME(0.0,49.0),PERC(0.0,100.0)
PAGE GROUP=(0.0,100.0)
OUTPUT TTIME(0.0,49.0),PERC1(0.0,100.0)
PAGE GROUP=(0.0,100.0)
OUTPUT TTIME(0.0,49.0),PERC2(0.0,100.0)
PAGE GROUP=(0.0,100.0)
OUTPUT TTIME(0.0,49.0),TOTAL,AUX
PAGE GROUP
OUTPUT TTIME(0.0,49.0),TOTAL1,AUXW
PAGE GROUP
OUTPUT TTIME(0.0,49.0),TOTAL2,AAUX
PAGE GROUP
OUTPUT TTIME(0.0,49.0),LOAD1,AUXWA
PAGE GROUP
OUTPUT TTIME(0.0,49.0),AUXWA(0.0,4000.0)
PAGE GROUP=(0.0,4000.0)
OUTPUT TTIME(0.0,49.0),LOAD
PAGE GROUP=(0.0,80000.0)
OUTPUT TTIME(0.0,49.0),AUXI (0.0,80000.0)
PAGE GROUP=(0.0,80000.0)
OUTPUT TTIME(0.0,49.0),LOAD,AUXI
PAGE GROUP=(0.0,80000.0)
OUTPUT TTIME(0.0,49.0),FP
OUTPUT TTIME(0.0,49.0),FR
OUTPUT TTIME(0.0,49.0),UL
TITLE SYSTEM DESCRIPTION
PRINT USE,PERC,MAXIN,LOAD,TOTAL,TI1,TRH,ALX,YY,XX,SOLAR,...
CO,TI2,TI3,TOUT,SB,CB,SF,CF,CC,TTE,DTI1,DTI2,DTI3,AUXIL3,...
AUXWA,AUXW,AAUX,TOTAL1,TOTAL2,PERC1,PERC2,IND,LOAD1,IHOT,...
TKK,TUU,TUUTKK,FFSS,HHUU,UI,COP,D4,M,CC,F,FP,FR,TWA,WAT,AUXI
TIMER OUTDEL=1.0,FINTIM=1176.0,DELT=0.25,PRDEL=1.0
METHOD RKSFX
END
```

APPENDIX C

Simulation Program of Cooling

```
*********************************************************************
*                                                                   *
*    SIMULATION PROGRAM FOR COOLING FOR MODEL A AND MODEL C        *
*                                                                   *
*********************************************************************
INITIAL
/       DIMENSION A(360),B(360),C(10)
CONST CP=12500.0,ZE=0.0,A2=7000.0,R1=1.0,A3=12000.0
FIXED I,J,K
INCON IC=55.0
NOSORT
     5 FORMAT(12F5.3)
     6 FORMAT (3F6.4)
       READ (5,5) (A(I),I=1,360)
       READ (5,5) (B(I),I=1,360)
       READ (5,6) (C(I),I=1,3)
DYNAMIC
NOSORT
       IF (TIME.EQ.ZE) GO TO 125
       GO TO 120
   125 MAXIN=0.0
       TA=75.0
       AAUUX=0.0
       TOTAL=0.0
       AUX=0.0
       TI1=IC
       TI2=IC
       TI3=IC
       SW1=0.0
       SWI1=0.0
       GO TO 150
   120 TE=TIME+0.99
       J=TE
       TA=B(J)
       ANG=TIME/360.0+1.0
       K=ANG
       CD=C(K)
       TTE=J
       AH=TTE*0.2618
       CH=SIN(AH)
       IF (CH.LE.0.) GO TO 150
       SB=0.777*CD*CH+0.629*(1.0-CD**2)**0.5
       CB=(1.0-SB**2)**0.5
       SF=CD*(1.0-CH**2)**0.5/CB
       CF=(1.0-SF**2)**0.5
       MAXIN=A(J)*3.68663/SB
       GO TO 130
   150 CB=0.0
       SB=0.0
       SF=0.0
       CF=0.0
       MAXIN=0.0
   130 CONTINUE
SORT
       HR=(4.090*CB*SF+2.556*CB*CF)*MAXIN
DHR=314.14*(TA-75.0)+HR
DWALL=DELAY(16,4.0,DHR)
       INFIL=326.1*(TA-75.0)
```

```
      WIND=(31.680*CB*SF+19.80*CB*CF)*MAXIN+200.0*(TA-75.0)
      KIT1=IMPULS(4.0 ,6.0)
      KIT2=IMPULS(4.25,6.0)
      KIT3=IMPULS(4.50,6.0)
      KIT4=IMPULS(4.75,6.0)
      KIT5=IMPULS(5.00,6.0)
      KITCH1=KIT1+KIT2+KIT3+KIT4+KIT5
      KIT6=IMPULS(16.00,24.0)
      KIT7=IMPULS(16.25,24.0)
      KIT8=IMPULS(16.50,24.0)
      KIT9=IMPULS(16.75,24.0)
      KIT10=IMPULS(17.0,24.0)
      KITCH2=KIT6+KIT7+KIT8+KIT9+KIT10
      LOAA=(CWALL+WIND+INFIL+1200.0*(KITCH1-KITCH2)+1350.0)*...
       1.3
      LOAD=INSW(LOAA,ZE,LOAA)
      HORI=MAXIN*SB
      SOLTH=MAXIN*CB*CF
      EAWF=MAXIN*CB*SF
      XX=INSW(LOAD,ZE,R1)
      TRB=TI3+LOAD/A2
      FUN=IMPULS(16.0,24.0)
NOSORT
      IICOP=0.53*(TA+460.0)/(TA-35.0)
      AAUX=LOAD/(IICOP-1)
      IF (TIME.EQ.ZE) GO TO 170
      IF (KEEP.EQ.0) GO TO 170
      TOTAL =TOTAL+LOAD*XX*DELT
      AAUUX=AAUUX+AAUX*DELT*XX
      IF(TI3.GE.55.0) GO TO 185
      SWII=0.0
  170 CONTINUE
      IF (FUN.EQ.1.0) GO TO 180
      IF (SWI.EQ.1.0) GO TO 180
      IF (SWII.EQ.1.0) GO TO 180
      YY=0.0
      COP=0.0
      GO TO 190
  185 SWII=1.0
  180 IF (TI1.LE.40.0) GO TO 190
      YY=1.0
      COP=0.53*(TA+460.0)/(TA-TI1+30.0)
      CAPA=7633.65*(COP-1)
      THB=TI1-CAPA/A3
      IF(TIME.EQ.ZE) GO TO 200
      IF(KEEP.EQ.0) GO TO 200
      AUXI=7633.65*DELT
      AUX=AUX+AUXI
  200 CONTINUE
      IF (TA.GE.60.0) GO TO 192
      SWI=1.0
      GO TO 191
  190 THB=TI1
  192 SWI=0.0
  191 CONTINUE
SORT
DTI1=(A2*(TRB-TI1)*XX+A3*(TI2-TI1)*YY+11.6*(75.0-TI1))/CP*3.0
TI1=INTGRL(IC,DTI1)
DTI2=(A2*(TI1-TI2)*XX+A3*(TI3-TI2)*YY+8.45*(75.0-TI2))/CP*3.0
```

```
TI2=INTGPL(IC,DTI2)
DTI3=(A2*(TI2-TI3)*XX+A3*(TAB-TI3)*YY+8.45*(75.0-TI2))/CP*3.C
TI3=INTGRL(IC,DTI3)
TTIME=TIME/24.0
PAGE XYPLOT,HEIGHT=5,WIDTH=7
OUTPUT TTIME(0.0,14.0),LOAD
OUTPUT TTIME(0.0,14.0),IICOP,COP
PAGE GROUP
OUTPUT TTIME(0.0,14.0),TI1,TI2,TI3
PAGE GROUP
OUTPUT TTIME(0.0,14.0),AUX,AAUUX,TOTAL
PAGE GROUP
OUTPUT TTIME(0.0,14.0),TI3
TITLE SYSTEM DISCRIPTION
PRINT TA,TOTAL,AUX,MAXIN,HORI,EAWE,SOUTH,CH,SB,CB,SF,CF,...
IICOP,COP,AUXI,AAUX,AAUUX,TI,SWI,SWII,CAPA,THB,LOAD,XX
TIMER OUTDEL=1.0 ,FINTIM=336.0,DELT=0.25  ,PRDEL=1.0
METHOD RKSFX
 END
```

APPENDIX D

Weather Data during the Simulation

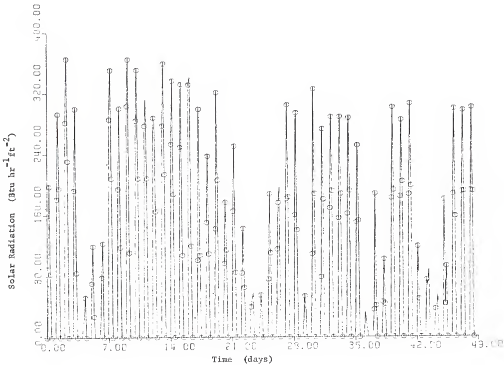

Figure A.1 Solar Radiation Impinging on the Collector during the Simulation.

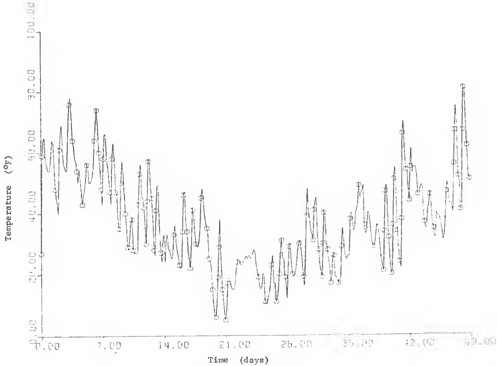

Figure A.2 Ambient Temperature Data during the Simulation.

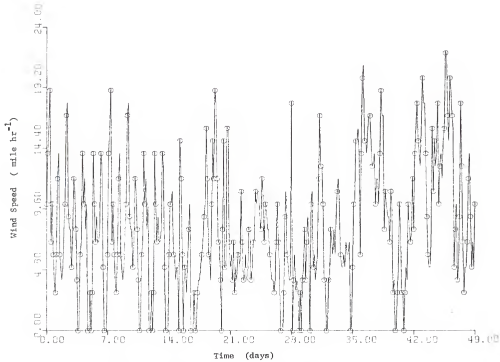

Figure A.3 Wind Speed Data during the Simulation.

SIMULATION OF SOLAR HEATING AND COOLING SYSTEMS USING THE CONTINUOUS SYSTEM MODELING PROGRAM

by

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B. S., Taiwan University, 1973

AN ABSTRACT OF A MASTER'S THESIS

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MASTER OF SCIENCE

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Manhattan, Kansas

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ABSTRACT

An economic feasibility study of direct use of solar energy for residential heating and cooling in Manhattan, Kansas has been made in this work. Three solar heating and cooling systems were modeled and simulated on a digital computer using the Continuous System Modeling Program. MODEL A, called SOLAR ENERGY HEATING AND CONVENTIONAL COOLING SYSTEM, was composed of a solar collector, an energy storage tank, an electric hot water heater, an electric furnace to supplement solar energy in the winter, a conventional air conditioner for air-conditioning in the summer, and the necessary pumps, piping, and controls. MODEL B, called SOLAR-ASSISTED HEAT PUMP FOR HEATING AND COOLING, was similar to MODEL A, with the exception that a heat pump was used in place of the air conditioner for house cooling. The heat pump was assumed to operate just like an air conditioner in the summer, but also was used for house heating in the winter. MODEL C, called SOLAR-ASSISTED HEAT PUMP FOR HEATING AND HEAT PUMP CHILLED WATER FOR COOLING, was equiped the same as MODEL B, except the heat pump cooling operation was designed to cool water in the storage tank during the summer night and the chilled water used to cool the house.

The simulations were made for a typical two story house located in Manhattan, Kansas. The hour-by-hour performance of each model was studied using the 1976-1977 hourly weather bureau data for the Manhattan area. Several collector areas were tested in these models and a cost analysis was used to determine the minimum cost system. The simulation results showed that the heat pump cooling operation in MODEL C was less efficient than in MODEL B, and that, for solar heating, the solar-assisted heat pump system (MODEL B) had a better thermal performance than the conventional

solar energy heating system (MODEL A). Under the assumed costs for solar equipment and other fees, the results of the cost analysis indicated that MODEL B was better than MODEL A, and that, MODEL B was economically competitive with a conventional all electric system at the present power rates. For a winter electrical rate of \$0.030/KWh in Manhattan, the results showed that the optimum collector size was 400 ft^2 for MODEL A and 600 ft^2 for MODEL B, and the annual dollar savings were about \$156 for MODEL B and an annual dollar loss of \$109 for MODEL A.